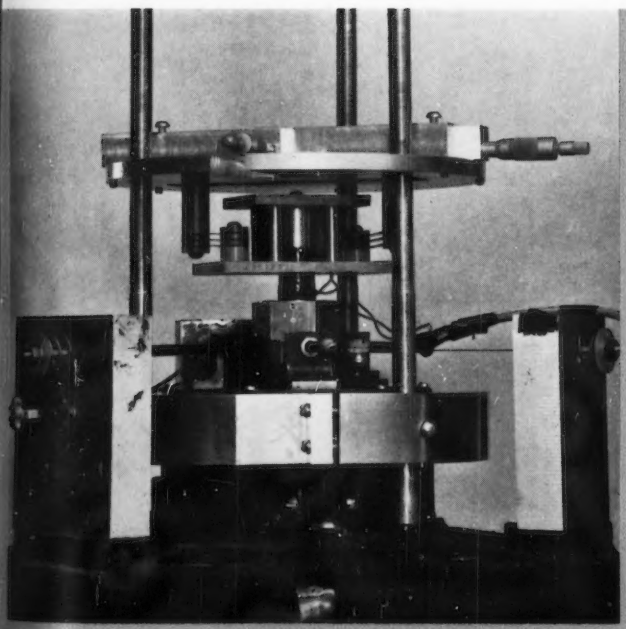
APRIL 1961

experimental mechanics



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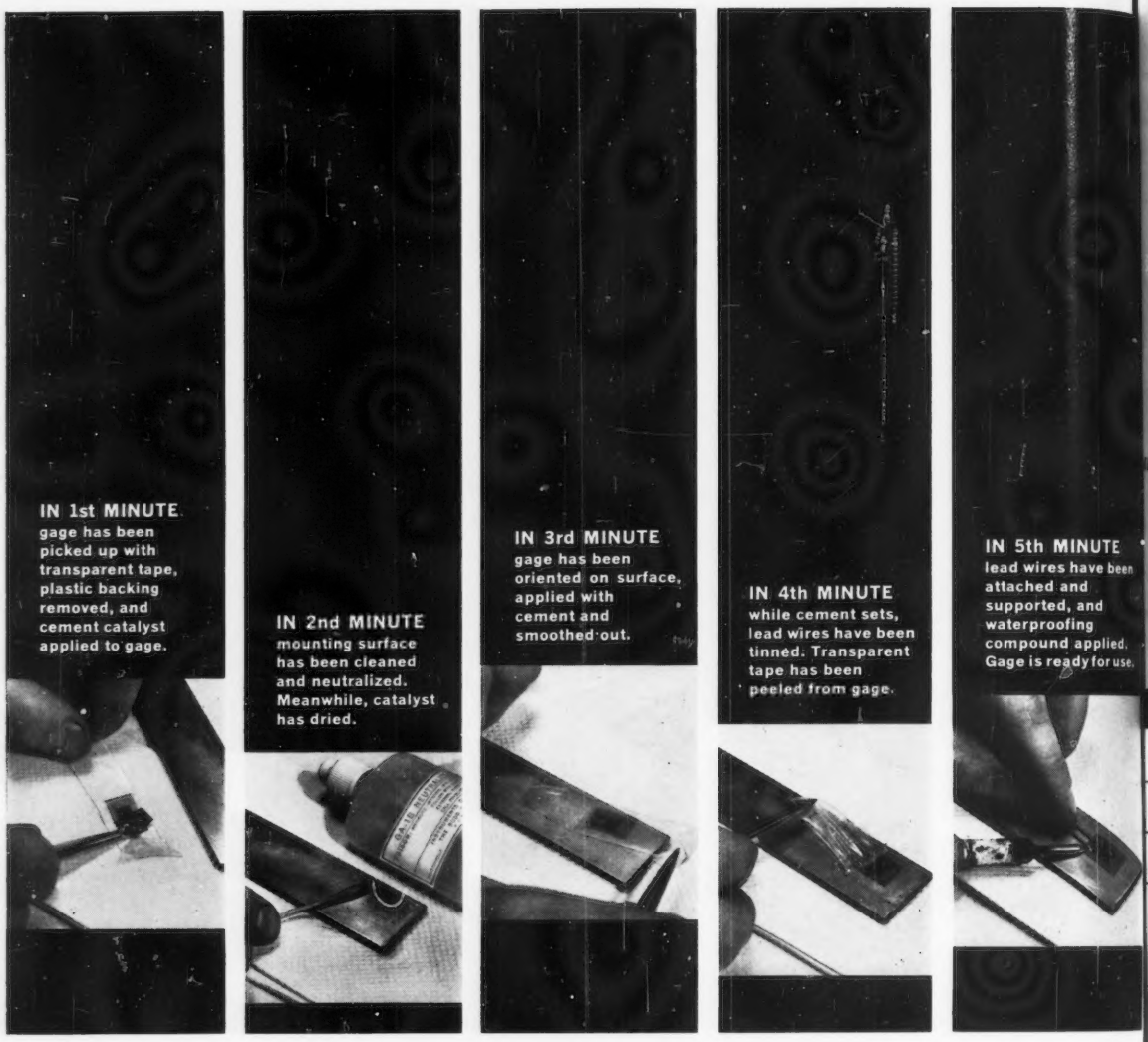
SESA JOURNAL OF THE SOCIETY FOR EXPERIMENTAL STRESS ANALYSIS


IN 1st MINUTE
gage has been picked up with transparent tape, plastic backing removed, and cement catalyst applied to gage.
IN 2nd MINUTE
mounting surface has been cleaned and neutralized. Meanwhile, catalyst has dried.
IN 3rd MINUTE
gage has been oriented on surface, applied with cement and smoothed out.
IN 4th MINUTE
while cement sets, lead wires have been tinned. Transparent tape has been peeled from gage.
IN 5th MINUTE
lead wires have been attached and supported, and waterproofing compound applied. Gage is ready for use.

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L OF THE SOCIETY FOR EXPERIMENTAL STRESS ANALYSIS

experimental mechanics

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SOCIETY FOR EXPERIMENTAL STRESS ANALYSIS
21 Bridge Square, Westport, Connecticut

SESA PUBLICATIONS

PROCEEDINGS OF THE SOCIETY FOR EXPERIMENTAL STRESS ANALYSIS

The PROCEEDINGS are a series of bound volumes comprised of the technical papers presented at National Meetings. These contain the results and descriptions of analytical and experimental investigations, as well as techniques for the measurement of mechanical strain. Published twice annually, they include up-to-date information on various phases of instrumentation. Currently available:

Volume III, No. 2
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Volume XV, No. 2
Volume XVI, No. 2
Volume XVII, No. 1
Volume XVII, No. 2

CUMULATIVE INDEX—PROCEEDINGS

Thirty-four books of the PROCEEDINGS have been published to date. This Index has been prepared to serve as an easy reference to any one of the 546 papers which have been published in the first thirty books. It consists of three parts: (1) chronological index, (2) author index, (3) topical index. It represents a "master reference file" to all PROCEEDINGS published, from Vol. I, No. 1 through Vol. XV, No. 2.

HANDBOOK OF EXPERIMENTAL STRESS ANALYSIS

This book opens with a thorough presentation of mechanical properties. It presents practical, usable information on: mechanical, electrical and optical gages, brittle coatings, photoelasticity, X-rays, service fractures, analogies, vibration measurements, theory of elasticity and precision of measurements. Stiff cloth binding—contains 1060 pages of text.

MANUAL ON EXPERIMENTAL STRESS ANALYSIS TECHNIQUES

For those who are new to the field of experimental stress analysis, the MANUAL introduces the fundamentals of experimental stress analysis in terms easily understood by the novice. It contains detailed instructions and, with the aid of illustrations and charts, elaborates on the application of three particular methods for determining stress: (1) SR-4 bonded wire strain gages, (2) stresscoat (brittle lacquer) and (3) photoelasticity.

STRAIN GAGE TECHNIQUES

Twenty-three chapters of lectures and laboratory experiments of special summer programs presented at leading universities, covering such topics as: fundamental concepts, selection of gages, basic considerations in strain-gage instrumentation, electrical and mechanical aspects of the gage system, the potentiometric circuit, the wheatstone bridge, amplitude modulation, fundamentals of measuring systems, some commercial systems and rosette analysis. Approximately 680 pages.

CATALOG OF EQUIPMENT USED IN EXPERIMENTAL STRESS ANALYSIS

The Catalog contains references to manufacturers of equipment, hardware and instrumentation used in the measurement of strain.

EXPERIMENTAL MECHANICS

Official monthly journal of the SESA. Includes technical papers and news items of interest to SESA members and to industry.

ORDER FORM

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Manual on Experimental Stress Analysis Techniques	Perry, CC and Bean, WT	2.00	1.25	3.00	Included	10¢			$
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Experimentally Speaking

SCOPE OF THE SESA

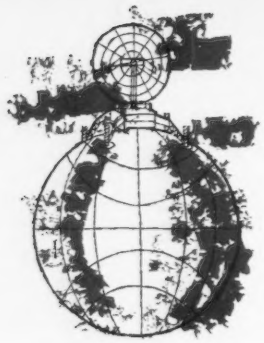
The first love of the Society for Experimental Stress Analysis was photoelasticity. Its second love was strain gages. As the Society grew, its interests encompassed broader fields. The first two loves did not exclude others even for the most enamored. They were the means by which some problems could be solved. To some the means of obtaining a solution was of more interest and importance than the solution itself, but most were more concerned with the solution and its consequences than with the technical problems involved in obtaining it. To show that there remained a considerable affection for the restricted area, and at the same time to indicate that this broad interest existed, the name "Society for Experimental Stress Analysis" was maintained and the title of the new journal was given as "Experimental Mechanics." The latter is more descriptive of the general interest of the Society.

Considering the papers presented at our national meetings in the past, the most numerous have dealt with descriptions of instruments, dynamic tests and stresses, techniques of application and use of strain gages, fatigue and photoelasticity. Many other subjects, of course, were also covered. The field of interest is quite broad, ranging from theoretical to practical studies, including human problems in biomechanics. Emphasis may vary; for instance, not long ago it appeared that photoelasticity had reached an age of maturity, and it was supposed that its application would become routine. But during the last few years, photoelasticity has experienced a renaissance of new techniques that makes it an indispensable tool for an increasing variety of problems.

There is always the possibility that the range of subjects may become so broad that interest in them may tend to become overly diluted. Our Society must and will grow as our science and technology progresses, but we must exercise care to see that the membership continues to be composed of those people who have a common field of interest and who can form a coherent group—and that the papers presented are within their scope of interest.

The scope of interest of the SESA is not limited to the techniques involved in experimental stress analysis. Generally, it includes experimental studies of the stresses, strains, responses and physical behavior of materials and structures under various types of mechanical excitation, together with instrumentation techniques involved. Our future may well lie within this broad scope—let us encourage its fullest development so as to provide the greatest value to our membership.

Irwin Vigness
VICE-PRESIDENT, SESA



tracking station

the industry in focus

Professional Income of Engineers

Engineers' salary levels rose approximately 5% per year between 1958 and 1960 according to a survey conducted by the Engineering Manpower Commission of Engineers Joint Council. The over-all median annual salary now stands at $9,600. The recent increase of 5% may be compared with an average annual increase of 6½% between the years 1953–1958. Seven years ago (1953), the first in this series of surveys found the median salary for engineers at $6,500. The survey covered approximately 200,000 engineering graduates in industry, education and government.

General Motors Sales Highest in 1960

General Motors' dollar sales in 1960 were the highest in the corporation's history and net income was second only to 1955 according to figures recently disclosed by F. G. Donner, chairman, and J. F. Gordon, president.

Unit sales of GM cars and trucks for 1960 also topped every other year except 1955, they announced. Dollar sales for the year 1960 are estimated at $12.7 billion, compared with $11.2 billion in 1959 and the previous record of $12.4 billion in 1955. Electro-Motive Div. is a corporate member of SESA.

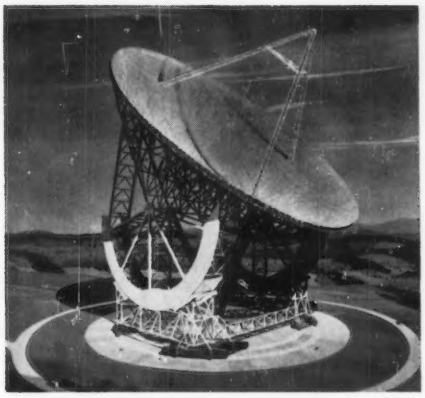
Datex Branch for Northern California

A new district sales and service office to serve the northern California area has been opened at 1885 The Alameda, San Jose by Datex Corp., Monrovia, Calif. George Cline, formerly sales engineer in the firm's Western Regional office, has been named manager. The new office is the first of several district and regional sales and service offices which Datex expects to open this year.

Column Strength at High Temperatures

A buckling metal column provides Battelle Institute with data to confirm the validity of a new analytical method for determining the compressive strength of columns at high temperature. The new method, based on short-time stress-strain properties of materials, predicts the stiffness of columns under conditions encountered by aircraft and missile structures. Previously, prediction of column properties was impossible when metal creep caused the shape to change continually.

Navy's "Big Dish" Radio-telescope



World's largest movable land object is this 600-ft "big dish" radio telescope now under construction at the Naval Radio Research Station, Sugar Grove, W. Va. Artist's sketch shows the 128 azimuth drive units and the 128 elevation drive units designed to have roll, pitch, yaw and translation motions to accommodate stresses from temperature changes, winds and icing conditions. The high-precision drives as well as the electronic control system is being built by Electric Boat Division of General Dynamics.

When the 30,000-ton telescope is completed, part of the operating time will be given to purely scientific studies such as radio astronomy, research in the characteristics of the earth's atmosphere, and gathering new geodetic and geomagnetic data on the earth itself. The remainder of the time will be given to advanced research in radio communications and navigation.

Babcock & Wilcox 1960 Earnings Highest in History

The Babcock & Wilcox Co. recently reported net income for 1960 of $17,818,000, or $2.88 per share, on shipments of $310,999,000. This income is 12% higher than that for the previous year and is the highest in the company's history.

Company officials point out that "these results were achieved, despite a 6% decline in shipments, primarily by continued general improvement in operating efficiency throughout the company." The Babcock & Wilcox Co. is a corporate member of SESA.

Application for Corporate Membership

SESA

Society for Experimental Stress Analysis

Name of Company _____

Name of Representative _____

Position or Title _____

Department _____

Number and Street _____

City _____ Zone _____ State _____

We wish to apply for Corporate Membership in the Society for Experimental Stress Analysis for the period indicated below:

_____ January 1, 1961 through June 30, 1961 Dues: $ 50.00

_____ January 1, 1961 through June 30, 1962 Dues: $150.00

We understand that the above amounts are based on fiscal-year dues of $100.00 and that we are to receive the full benefits prescribed by the Society for the designated period.

Date _____ Authorized Signature _____

_____ Title _____

(Please give billing address, if different from above. Mail completed application to:
Executive Secretary, Society for Experimental Stress Analysis, 21 Bridge Square, Westport, Conn.)

FUTURE MEETINGS

SESA

MAY 10–12. Spring Meeting. Benjamin Franklin Hotel, Philadelphia, Pa.

NOVEMBER 1–3. First International Congress on Experimental Mechanics. Hotel New Yorker, New York, N. Y.

•Related Conference:

OCTOBER 29–31. International Symposium on Photoelasticity. Illinois Institute of Technology, Chicago, Ill.

ARS

APRIL 5–7. Conference of the American Rocket Society. Lifting Re-entry Vehicles: Structures, Materials, Design. Palm Springs, Calif.

ASME

JUNE 14–16. Conference of American Society of Mechanical Engineers. Applied Mechanics Conference. Illinois Institute of Technology, Chicago, Ill.

ASTM

JUNE 25–30. Annual Meeting of American Society for Testing Materials. Chalfonte-Haddon Hall, Atlantic City, N. J.

IES

APRIL 5–7. Annual Technical Meeting and Equipment Exposition. Institute of Environmental Sciences. Park Sheraton Hotel, Washington, D. C.

ISA

APRIL 17–19. Symposium on Instrumental Methods of Analysis. Instrument Society of America. Shamrock-Hilton Hotel, Houston, Tex.

Helpful McGraw-Hill books for your professional library

THEORY OF PLATES AND SHELLS

Covers the general theory of bending of plates under lateral load, with applications to circular and rectangular plates, and with numerous tables simplifying the calculation of deflections and stresses in plates. Provides a treatment of combined bending and tension or compression of plates, with applications to thin plates. **By S. Timoshenko, Stanford Univ.; and S. E. Woinowsky-Krieger, Université Laval, Quebec, Canada, 2nd Ed., 580 pp., 280 illus. $15.00**

THEORY OF ELASTICITY

For engineers concerned with calculating stresses in engineering structures and machine parts—here is essential knowledge of the theory of elasticity, together with the solution of engineering problems of practical importance. The book discusses approximate and experimental methods of solving elastic problems which have proved useful in the study of complicated cases of stress distribution. **By S. Timoshenko and J. N. Goodier, Stanford Univ., 2nd Ed., 506 pp., 266 illus., $11.50**

FORMULAS FOR STRESS AND STRAIN

A compact and readily usable summary of important formulas, facts, and principles pertaining to strength of materials. This reliable reference for engineers provides helpful formulas for stress analysis and elasticity. All experimental data and empirical formulas are in line with current information. New material has been added, ensuring complete coverage of stress and strain. **By Raymond J. Roark, Univ. of Wisconsin, 3rd Ed., 381 pp., $8.25**

THE STRAIN GAGE PRIMER

What the bonded wire resistance strain gage can do in solving problems in experimental stress analysis is explained in this practical book. All phases are covered—from selecting the proper commercial gage through preparing surface of the test member, cementing, damage-proofing, and wiring the gage —to selecting the proper instrument to record the strain gage data, and interpreting readings in terms of significant stresses. **By C. C. Perry and H. R. Lissner, Wayne Univ., 280 pp., 180 illus., $6.00**

THE TESTING AND INSPECTION OF ENGINEERING MATERIALS

A practical approach to the principles and methods of materials testing, with full details on types of tests in use. Describes modern testing equipment and covers effects of variables on test results. Includes information on micro-hardness testers, non-destructive tests, low-temperature impact tests, and more. **By H. E. Davis, G. E. Troxell, Univ. of Calif.; and C. T. Wiskocil, formerly, Univ. of Calif. 2nd Ed., 431 pp., 171 illus., 42 tables, $7.50**

APPLIED ENGINEERING MECHANICS

Emphasizes the use of reasoning rather than complicated mathematics for solving engineering problems. Employing only simple mathematics, the book proceeds from easy to difficult phases of engineering mechanics, clarifying the principles advanced with numerous illustrations. Takes you step-by-step through the solutions of scores of technical problems. **By Alfred Jensen, formerly, Univ. of Washington; assisted by Harry H. Chanoweth, Univ. of Washington. 2nd Ed., 409 pp., 746 illus., $6.50**

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personnel
in the news

KRUMHANSL TO HEAD CORNELL LABORATORY

James A. Krumhansl, professor of physics at Cornell University, has been appointed director of Cornell's Laboratory of Atomic and Solid State Physics, succeeding Prof. Robert L. Sproull, recently appointed director of the Cornell Materials Science Center.

As head of the Laboratory, Prof. Krumhansl will direct and coordinate solid state research in physics and engineering physics.

Prof. Krumhansl has been associated with physics, electrical engineering and research administration in both industry and higher education. He has been involved in research on pulse communication, microwaves, network theory, analog computers, operations research, applied mathematics, theoretical solid state physics, chemical physics and industrial research planning.

A native of Cleveland, Prof. Krumhansl received his B.S. degree from the University of Dayton, M.S. from Case Institute of Technology and Ph.D. from Cornell.

CHIPMAN BECOMES MANAGER

The appointment of Robert D. Chipman as product manager for strain gages was announced recently by Harry Stern, president of Allegany Instrument Co., Division of Textron Electronics, Inc., Los Angeles, Calif.

Mr. Chipman's background and experience covers a broad area in the fields of theoretical and experimental mechanics as well as customer-oriented engineering.

OWENS MADE MANAGER

The appointment of William D. Owens as general manager of the Heiland Division of Minneapolis-Honeywell Regulator Co. has been announced by Paul B. Wishart, Honeywell president.

He succeeds S. A. Keller, who has resigned to accept a position with another company. Mr. Owens has been with Honeywell 10 years and, since 1958, has been manager of the company's Ordnance Division operations in Duarte, Calif.

The Heiland Division, with headquarters in Denver, Colo., is a producer of photographic equipment and high-speed direct-reading oscillographs.

PIQUE ELECTED VICE PRESIDENT

The board of directors of Statham Instruments, Inc., have elected Z. W. Pique as vice president—marketing, L. D. Statham, president, recently announced. Mr. Pique brings to Statham a background of 30 years of marketing and sales management experience.

He has been associated with Hoffman Electronics as vice president—marketing; Hughes Aircraft as director of marketing, Hughes Product Group; Texas Instruments as general manager, Capacitor Department; Westinghouse as district apparatus and supplies sales manager.

A native of Kentucky, Mr. Pique received a B.S. M.E. from the University of Kentucky and a B.A. in Humanities from Transylvania. He has been a registered electrical and mechanical engineer since 1938.

FINKELSTEIN AT BATTELLE

Theodor Finkelstein has joined the mechanical engineering research staff of Battelle Memorial Institute where he will head studies on Stirling-cycle engines and magnetohydrodynamics. He comes to Battelle from the University of Wisconsin where he served as assistant professor of mechanical engineering research.

Dr. Finkelstein received his doctorate from Imperial College, London. Before coming to the United States in 1958, he held the post of assistant chief scientific officer for the English Electric Co.

CAIRNS MADE DIVISIONAL MANAGER

William F. Cairns was recently appointed manager of the Cambridge Division of Airpax Electronics Inc. The plant is located in Cambridge, Md. Responsibilities include all phases of engineering, production and sales of the electromechanical and magnetic components produced at the Cambridge, Md., plant.

Prior to joining Airpax, Mr. Cairns was assistant to the president of the Chicago Steel and Wire Co.

He holds B.S. degrees in business administration (Cum Laude), mechanical engineering and aeronautical engineering, from the University of Pennsylvania.

McCANN APPOINTED MANAGER

Kearfott Division, General Precision, Inc., announced the appointment of F. D. McCann as manager of its New York Industrial Sales Office. Mr. McCann was associated with the Hertner Electric Co., a Kearfott subsidiary, for 15 years, and more recently he has been a representative for Kearfott power equipment and ultrasonic products. The new office will continue to handle all Hertner products as well as these two Kearfott special product lines.

Assisting Mr. McCann in servicing customers in the southern New York and New Jersey area will be sales engineers Richard Cunningham and John Grech.

STARR JOINS ALLEGANY

James E. Starr, formerly of Budd Instruments Division, has joined the Allegany Instrument Co., Division of Textron Electronics, Inc., as research director. Mr. Starr's background includes specialization in strain gage technology. He is located at the Alinco plant in Cumberland, Md., where he is directing the newly expanded research facilities formerly headed by G.William Harrison, now general manager. Mr. Starr will conduct research primarily on transducers, related electronics and strain gages.

James E. Starr

METER-INDICATOR

Bytrex Corp., 50 Hunt St., Newton 58, Mass., announces the development of Model 470, an instrument designed to provide continuous indication of pressure, torque, force, weight or flow when used in conjunction with strain-gage transducers of the bonded or unbonded type.

Model 470 is packaged complete with transistor amplifier, power supply and wide scale meter eliminating the necessity for any additional instrumentation, the manufacturer reports.

The unit was developed for use with strain-gage torque pickups to measure torque requirements on gas-turbine-engine accessories.

The Model 470 is available with a wide variety of features for adaptation to any special purpose requirement. Scale can be graduated to indicate any measured variable directly.

Complete details are given in Bulletin 103-A. **51**

HIGH STABILITY POWER SUPPLY

High stability current-regulated power supplies eliminating the need for using motor generator systems to maintain precise current control are announced by Automation Industries, Inc., Magnetics Division, 1090 Mills Way, Redwood City, Calif. The power supplies as described, utilize two separate control circuits which function independently, although both are completely contained in the package. The control circuits are said to be self-adjusting and compensate for both slow and fast variations resulting from dynamic input or output changes. The desired current output is obtained by simple adjustment of a single high-precision potentiometer control. The current output thus selected will maintain regulation of better than 0.02%, the manufacturer states. **53**

MICROMIDGET CHOPPER

According to Airpax Electronics, Inc., Cambridge, Md., Model 36 chopper employs a center-tapped drive coil to facilitate transistor drive in totally transistorized circuits. Electrical noise is brought to an irreducible minimum; even at high impedances the noise is down in the random noise level.

Model 36 is said to operate continuously over a temperature range of −65 C to +100 C. This chopper meets specifications during and after vibrations in any direction equivalent to sinusoidal vibration at an amplitude of 0.060 in. total travel over a frequency of 10 to 55 cps and 15 g from 55 to 2500 cps. The contacts are suitable for handling input signals up to 10 v dc at a maximum current of 2 ma, it is stated. **55**

PULSE GENERATOR

The Steinwood Tri-Pulse generator is now available from the Harwood Co., 1141 W. Valley Blvd., Albambra, Calif.

Specifically designed to provide a common time base for guidance of data-reduction personnel, the new generator feeds 10, 100 or 1000 pulses per second to as many as ten cameras or telemetering stations, the manufacturer reports.

It weighs less than 1½ lb, and power is supplied by 30-v battery. The new generator is said to have performed within 1% of selected frequency under the severe environmental conditions of sled runs exceeding 40 G's.

In addition to rocket-sled tests, current applications include remote camera operation in weapons research—missile firings, airborne aircraft, torpedo research work—and as a laboratory time standard for checking other instruments. **56**

new products

For more details regarding these products, circle item number on Reader Information Card.

LINEAR ACCELEROMETER

Small size, light weight and high-frequency response over a wide temperature interval are said to be the prime advantages of the new Model A507TC unbonded strain-gage linear accelerometer, which has been introduced by Statham Instruments, Inc., 12401 W. Olympic Blvd., Los Angeles 64, Calif.

According to manufacturer, the Statham A507TC unit is temperature compensated and gas damped, reducing the effects of temperature change to a minimum.

Ranges of the A507TC are from ±25 g through ±100 g. The unit provides relatively constant damping from −65° to +250° F, without requiring a heater jacket. Excitation is 5 v dc or ac (rms) through carrier frequencies. Full-scale open-circuit output is approximately ±20 mv at 5 v. Weight is approximately 4 oz. **52**

TEMPERATURE-INDICATING COATINGS

Tempilaq° temperature-indicating coatings are now available in aerosol spray cans from Tempil° Corp., 132 W. 22nd St., New York 11, N. Y. Forty systematically spaced temperature ratings covering the range from 100° F (38° C) to 650° F (343° C) inclusive, can be obtained in a choice of the spray-type container or the standard glass bottles equipped with applicator brushes. Development work is under way to perfect aerosol packaging of the 40 higher temperature ratings of Tempilaq° spanning the range from 700° F (371° C) to 2500° F (1371° C), which presently are supplied in the standard packaging for brush application only. The spray-can package has been developed to meet the need for a convenient method of coating large surfaces with Tempilaq°. 57

D-C AMPLIFIERS

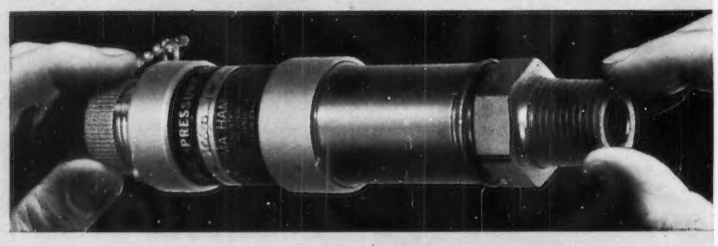
Two chopper-stabilized d-c amplifiers, models 602 and 98, have recently been developed by Video Instruments Co., Inc., 3002 Pennsylvania Ave., Santa Monica, Calif.

Model 602 is especially designed to amplify signals from strain gages, thermocouples and other transducers which provide low-level signals.

According to the manufacturer, the most distinguishing feature of the 602 is its output capability. It can deliver 200 ma into a 50 ohm load, and can be loaded capacitively up to 10 microfarads without going into oscillations, utilizing the full 100 cps bandwidth.

Model 98 is a subminiature signal conditioning d-c amplifier with a low-level solid-state chopper. The manufacturer states that this amplifier is designed for use with strain gages, pressure transducers, accelerometers, resistance probes, and thermocouples where compactness and operations under severe environmental conditions are important. 58

NEW GP PRESSURE CELL BY BLH ...NEVER BEFORE SUCH ACCURACY AND STABILITY AT SUCH LOW COST



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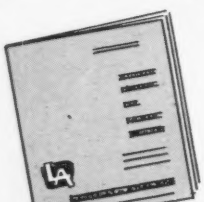

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MINIATURIZED SIGNAL AMPLIFIER

A miniaturized signal amplifier, CA17, with a size of 4 cu in. and a weight of 4.25 oz has been introduced by Statham Instruments, Inc., 12401 W. Olympic Blvd., Los Angeles 64, Calif. The tiny, transistorized CA17 provides for total utilization of low-level signals and proves itself more reliable in adverse environments than larger and heavier amplifiers, according to the manufacturer.

The Statham CA17 may be used with any resistance strain-gage transducer. Potentiometers are provided to permit zero and sensitivity adjustments. The signal output of the carrier amplifier is isolated from the power supply input.

It is reported that the output of the CA17 is −0.5 to +5 v d-c, with a power requirement of 35 ma at 28 v dc ±5%. Frequency response is flat ±4% (referenced to dc) from 0–2000 cps. The CA17 is designed to operate with transducers with a rated sensitivity from 1.5 mv per volt to 10 mv per volt. **59**

DATA-PROCESSING SYSTEM

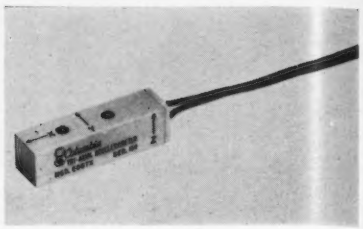
Burroughs Corp., Detroit 32, Mich., announces that it has developed a new solid-state electronic data-processing system designed to solve complex business and scientific problems.

A medium-priced system, the B5000 will provide performance equal to or greater than so-called large-scale computers, according to the manufacturer.

It is said to depart from traditional concepts of computer design to drastically cut programming time and problem-solving costs. **60**

ULTRAMINIATURE ACCELEROMETER

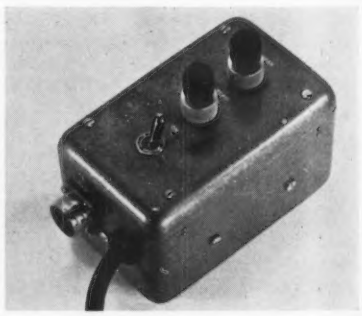
The announcement of the development of the Model 606-TX accelerometer, an ultraminiature sensing device for use in shock and vibration testing of printed circuit boards, electron tubes, sheet-metal structures, internal bores and other similar type



applications where mass loading is prohibitive or space requirements severely limited, has just been made by Columbia Research Laboratories, MacDade Blvd. & Bullens Lane, Woodlyn, Pa.

The Model 606-TX has a sensitivity of 2.0 mv/g along the X, Y and Z axis and a flat frequency response from 0.5 cp to 8 kc within ±5%. Resonant frequency is 50 kc. Open circuit sensitivity is 5.5 mv/g, maximum acceleration is 1000 g and amplitude linearity is ±2%. Temperature range of the Model 606-TX is from −65° F to +350° F for standard units and −65° F to +500° F for high-temperature units, with less than ±10% variation in sensitivity over both ranges.

The case of the accelerometer is stainless steel and measures 0.4 x 0.4 x 1.5 in. The weight of the unit is just 8 g. **61**



CARRIER-DEMODULATOR

Pace Engineering Co., 13035 Saticoy St., North Hollywood, Calif., announces a miniature carrier-demodulator for use with Pace variable-reluctance pressure transducers, providing a d-c output signal for recording and control in d-c systems.

Small enough to be included in the case of most receiving instruments, the Model CD10 carrier-demodulator is designed for operation on 95–125 v, 60 cps at 5 w, the manufacturer reports. Regulated against input voltage variation, the new instrument's output is 1 ma into a 1000 ohm load.

The Model CD10 carrier-demodulator is said to operate reliably over an ambient temperature range of −65 to +165° F. **62**

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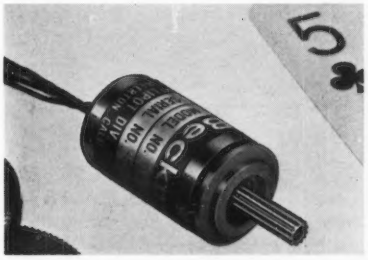
RADIANT HEATERS

Portable glass radiant heaters said to be most efficient for curing and welding of plastic parts are offered by Plastic Weld Co., 651 Snoqualmie St., Seattle 8, Wash. Available in a range of sizes from small panels to enclosed ovens, the heaters are said to afford maximum efficiency and uniformity within close tolerances.

For higher temperatures, suitable for stress relieving, the company also offers a line of high-intensity ceramic quartz heating units using short-wavelength energy. Temperatures from ambient to 3500° F are attained and controls are said to hold temperatures within ±2.5° F. **63**



SIZE 5 SERVOMOTOR

Helipot Division of Beckman Instruments, Inc., 2500 Fullerton Rd., Fullerton, Calif., announces initial production of Size 5 servomotors.

The Size 5 servomotor, designated model 9005-1502-0, is described as 0.865 in. long and 0.6 oz in weight. Powered by 26-v, 400-cycle reference voltage, the servomotor has a no-load-speed of 10,000 rpm. Torque-at-stall registers 0.1 oz in., while rotor inertia is 0.18 g cm², providing an acceleration-at-stall of 39,000 rad/sec². **65**

DIGITAL VOLTMETERS-RATIOMETERS

Non-Linear Systems, Inc., Del Mar, Calif., has broadened the applications for its Series 30 medium-

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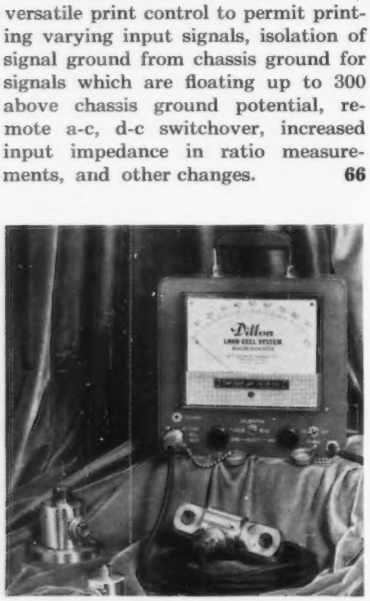
priced digital volt meters-ratiometers, with announcement of the new four-digit V34A and five-digit V35A.

The manufacturer reports that both instruments incorporate a number of new features while retaining transistorized "no-needless-nines" logic, plug-in oil bath stepping switches and other basic features of the V34 and V35 which they replace in the NLS line.

The new features include a more versatile print control to permit printing varying input signals, isolation of signal ground from chassis ground for signals which are floating up to 300 above chassis ground potential, remote a-c, d-c switchover, increased input impedance in ratio measurements, and other changes. **66**



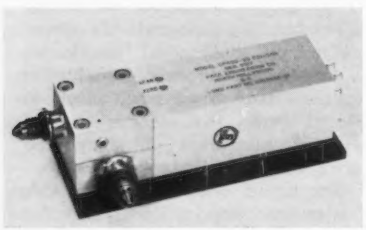
REMOTE-READING LOAD-CELL SYSTEM

The availability of a new remote-indicating load-cell system which permits instantaneous, direct readings of forces, in pounds, at distances over 1000 ft from the actual cell, is announced by W. C. Dillon & Co., Inc., 14620 Keswick St., Van Nuys, Calif. Applications are said to be found wherever site proximity is inconvenient, dangerous, confined or outside of naked-eye range.

The manufacturer reports that typical uses of this are on cranes and hoists; electronic scales; measuring structural stresses in wind tunnels; measuring jet engine thrusts; adjusting and maintaining proper tension in catapult and arresting gear on aircraft carriers.

This load-cell system utilizes a differential-transformer principle, whereby all tubes, amplifiers and circuitry has been eliminated by a special inverse-ratio circuit which carries sufficient voltage for integral magnification. The temperature compensated circuit utilizes semiconductors entirely. Separate reference charts, conversion tables and circuit balancing are also completely eliminated.

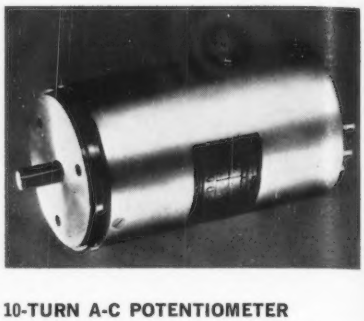
A 12-page, profusely illustrated brochure is available upon request. **67**



PRESSURE TRANSDUCER

A d-c output variable-reluctance pressure transducer for voltage-controlled telemetering and other d-c systems is now in production at Pace Engineering Co., 13035 Saticoy St., North Hollywood, Calif.

According to the manufacturer, Model CP49 combines, in a single package, a variable-reluctance pressure transducer and a d-c energized carrier-demodulator to realize both the operational simplicity of d-c output and the performance advantages of magnetic reluctance transducers: frequency response, long life, vibration, shock and overload resistance. Gage, differential and absolute models are available with ranges from 5 to 10,000 psi. Overpressure is 700 psi or 200% of range, whichever is greater and zero shift with line pressure is less than 1% full scale. **68**



10-TURN A-C POTENTIOMETER

Helipot Division of Beckman Instruments, Inc., 2500 Fullerton Rd., Fullerton, Calif., announces production of a series of 10-turn precision potentiometers for a-c excited circuits.

The 7800 Series A-C pot, a 2-in. diameter unit, is reported by the manufacturer to be a major improvement over the use of conventional potentiometers in a-c applications. It has high input impedance, and low output impedance, thereby substantially reducing quadrature and loading effects.

Standard model 7803 has an impedance range of 1000 to 200,000 ohms and covers a frequency range of 400 to 1000 cps. **69**

COOLED LINEAR AMPLIFIER

A high-frequency linear power amplifier that delivers 1000-w power and accepts AM and single side band voice, digital data link or other type of modulated signal in the 2–36 mc frequency range is being manufactured by Hughes Aircraft Co., P. O. Box 90–902, Los Angeles 45, Calif.

The new power amplifier is des-

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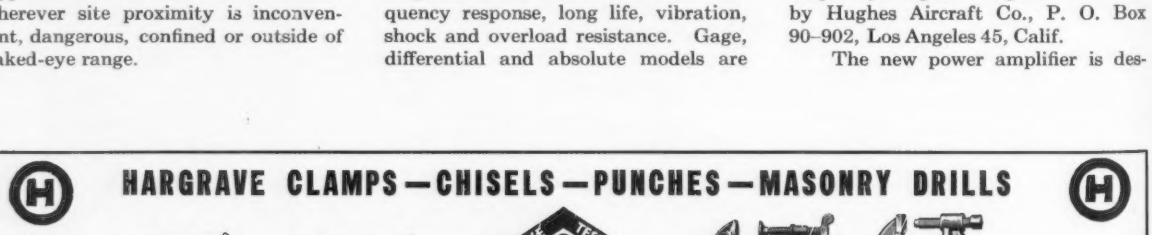

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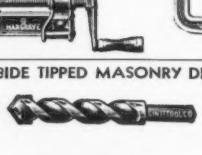

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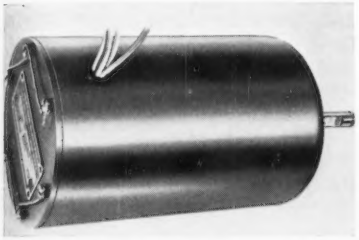
on Page 17A

ignated HC-105. Because of a new cooling technique, its rf circuits, servo tuning controls, high-voltage power supply, filament supply, drive stage and output stage are enclosed in a $7^1/_2$ by $7^1/_2$ by $6^1/_2$-in. case, the manufacturer states.

The amplifier is installed within a finned case filled with a liquid fluorochemical. Approximately 1600 w of heat are said to be dissipated, and mean temperature within the case is kept below 160° C. In addition to its cooling action, the fluid acts as an insulator. **70**



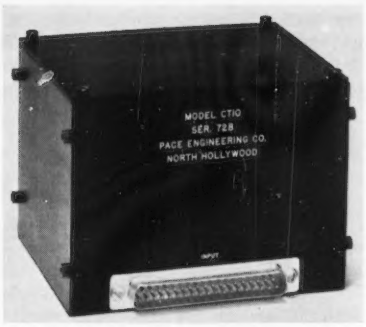
AXIAL VANE-BLOWER INDUCTION MOTOR

Kearfott Division of General Precision, Inc., 1150 McBride Ave., Little Falls, N. J., has announced the addition of the DE-30-1 unit to its line of miniature and subminiature special service motors. These motors are manufactured by the Hertner Electric Co., a wholly owned subsidiary of Kearfott. This induction motor is specifically designed to drive an axial vane blower utilized in such applications as cooling electronic equipment and crew compartments. When equipped with the proper blower the unit can be used in altitudes up to 30,000 ft, and Class H high-temperature insulation incorporated into the motor permits operation in temperatures up to 180° C. **71**

TELEMETERING SUB-CARRIER OSCILLATOR SYSTEM

Pace Engineering Co., 13035 Saticoy St., North Hollywood, Calif., announces the production of the model CT10 telemetering sub-carrier oscillator system that provides a compact and reliable telemetering unit for operation with up to 12 variable inductance transducers.

The instrument's module case accepts six printed circuit sub-carrier oscillator cards, which together with appropriate transducers may be tuned



to any of the IRIG bands, 4 through 18 or optional bands, A through E. Included, also, is a dual channel composite amplifier that receives the summed output frequencies from the inductance oscillators and feeds the transmitter.

Weighing less than 2 lb, the Model CT10 is only 3 x 3 x $3^3/_4$ in. It is said to withstand vibration of 35 g, 50–2000 cps; shock and acceleration of 200 g, any axis, with less than 1% bandwidth change. **72**

LOW-FREQUENCY ADAPTER

In order to extend the range of present ultrasonic inspection systems, a new instrument designated "low-frequency adapter" has been developed by Automation Industries, 3613 Aviation Blvd., Manhattan Beach, Calif. This instrument, when used in conjunction with standard pulse-echo ultrasonic equipment, provides ultrasonic inspection down to 200 kc, according to the manufacturer.

It is also reported that materials previously opaque to ultrasound, such as plastics, refrasil, castings, rubber and other high attenuating materials, can now be penetrated with ultrasound. Resolution and sensitivity are exceptionally good at these frequencies. Over 24 in. of cast Inconel X has been successfully inspected with the new instrument. **73**

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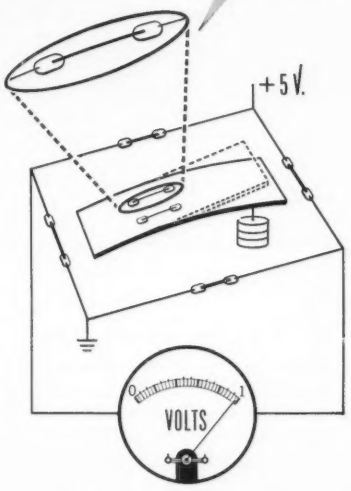

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ASA Acoustical Terminology

The American Standards Association, 10 E. 40th St., New York 16, N. Y., has published the newly developed "American Standard Acoustical Terminology (including Mechanical Shock and Vibration), S1.1—1960."

The standard contains 50 pages of definitions and tables grouped under 15 section headings. The 12-page index is cross-referenced. Price $4.50 per copy.

Standards Handbook

Precision Measurement and Calibration, National Bureau of Standards Handbook 77, issued Feb. 1, 1961, three volumes: Vol. I, *Electricity and Electronics,* 845 pp., $6; Vol. II, *Heat and Mechanics,* 965 pp., $6.75; Vol. III, *Optics, Metrology, and Radiation,* 1025 pp., $7 (order from Superintendent of Documents, U. S. Government Printing Office, Washington 25, D. C.).

This Handbook is a compilation of the more important National Bureau of Standards publications over a period of years, dealing with precision measurement and the calibration of standards. This new compilation is intended to serve as a quick reference source for workers in the field of standards, and also as a textbook and aid to scientists and engineers in standards laboratories.

Foil-strain-gage-Data

An 8-page brochure with a 1-page data insert cites the experience gained by Allegany Instrument Co., Division of Textron Electronics, Inc., Cumberland, Md., in developing their line of bonded resistance strain gages over the past 15 years.

Catalog EA-1 discusses in detail the various characteristics of these precision foil gages, qualitatively and quantitatively. Charts, tables and data sheet give physical properties and technical data. **105**

Polyester Strain Gage

Three bulletins on the polyester gage manufactured by Tokyo Sokki Kenkyujo Co., Ltd., of Tokyo, Japan, are available through H. M. Newhall & Co., 417 Market St., San Francisco 5, Calif. The first of these deals with "Direction for Use of Polyester Gauge"; the second is entitled "Polyester Gauge"; and the third describes the "S-1A Static Strain Meter."

The polyester gage is a resistance-wire strain gage composed of a grid of resistance wire and a supporting base impregnated with polyester resin. It is said to be applicable not only to metals but to concrete, woods and synthetic resins. **107**

Flexible Silicon Strain Gages

A technical paper on Recent Developments in Flexible Silicon Strain Gages, authored by William V. Wright of Electro-Optical Systems, Inc., and J. C. Sanchez of Micro Systems, Inc., is now available from Micro Systems Inc., 319 Agostino Rd., San Gabriel, Calif.

The paper discusses the properties of semiconductor silicon strain gages and stresses high gage factors coupled with reasonable temperature stability. Applications of the semiconductor devices in aircraft instrumentation and physiological measurements are presented. **109**

Accelerometers

Technical bulletin T-105, a short-form product listing which illustrates and describes a line of true compression-type accelerometers that have been designed for special and unusual applications in the missile and airborne vehicle shock and vibration field, has just been published by Columbia Research Laboratories, MacDade Blvd. & Bullen Lane, Woodlyn, Pa.

The bulletin describes eight different models of accelerometers and a calibration reticle which was developed for quick, on-the-spot calibration of vibration equipment. The accelerometers include three tri-axial units for measuring acceleration in mutually perpendicular X, Y and Z planes; a self-calibrating accelerometer for remote system checkout; one high-frequency and two high-sensitivity accelerometers; and a miniature rugged accelerometer whose output connector can be rotated to any convenient direction in the plane perpendicular to its sensitive axis. The calibration reticle is a simple visual device that allows direct calibration over a 30 to 200 cps range. **111**

Precise Angle Indicator

A 4-page bulletin giving circuit diagrams, specifications and performance characteristics of their Precise Angle Indicator CO2721011 is available from Kearfott Division, General Precision, Inc., Little Falls, N. J. The unit is a small servo device designed to provide direct angular readout of remote shaft positions. High accuracy is claimed for numerous applications ranging from computer trouble shooting to aircraft heading indications. **113**

Resistance Bridges and Power Supplies

A "Special Product Note" on resistance bridges and power supplies designed for use in airborne and ground-based applications with Trans-Sonics temperature transducers is now available from Trans-Sonics, Inc., P. O. Box 328, Lexington 73, Mass.

This Note also describes a completely integrated package for multi-point temperature-measuring systems. **115**

Application Stories on Transducer Systems

A series of articles on varied applications of high-accuracy transducer systems has been collected and released by Gilmore Industries, Inc., 13015 Woodland Ave., Cleveland 20, Ohio.

Included in the 42-page Bulletin TD-105 are such diverse stories as electronic weigh loading of hot metal, and pelletized ore; electronic weight, force and thrust measuring on aircraft and missiles; strain-gage testing of the B-52G and USS Albacore submarine; calibration of transducers used in aero space and temperature testing to detect machinery parts failure. **117**

Size 5 Servo Motor

A 2-page data sheet giving electrical and mechanical characteristics and performance charts for their size 5 servo motor is offered by Kearfott Division, General Precision, Inc., Little Falls, N. J. Designation is J126-06. **119**

You'll find
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on Page 17A

Transducers

Clark Electronic Laboratories, Palm Springs, Calif., has a condensed descriptive catalog "Micro-ducer News" featuring details on various types of solid state transducers. Described are miniature-force, pressure-, strain-, magnetic- and altitude-sensing transducers in various forms, as well as the company's line of pressure-sensitive paint. **121**

Six-component Thrust Stand

A new symmetrical six-component rocket-engine thrust stand developed by Wiancko Aeronautics, a division of Wiancko Engineering Co., 255 N. Halstead, Pasadena, Calif., is said to measure thrusts with $\pm 0.1\%$ accuracy. According to the manufacturer, the symmetrical arrangement of this stand virtually eliminates all cross-coupling effects during load transients, while improved structural stiffness increases over-all linear frequency response. For installations in restricted spaces, the symmetrical stand offers a size advantage.

A detailed description of this stand is contained in a 33-page Wiancko Brochure, WEC-BA-7A, "The Design of High-Accuracy Thrust Stands and Calibrators." **123**

Resistance Bridges

Synchro bridge KT427775-2A and resolver bridge KT427771-2A are described as to features and specifications in a data sheet offered by Kearfott Division, General Precision, Inc., Little Falls, N. J. Diagrams of test circuits are also given. **125**

Ceramic Cement

A special product note on a new ceramic cement is now available from Trans-Sonics, Inc., P. O. Box 328, Lexington 73, Mass., designers and manufacturers of precision pressure and temperature transducers.

This note describes ceramic cement, Type 64CP, newly developed by Trans-Sonics for bonding temperature or strain gage transducers to metallic or nonmetalic surfaces. Properties of the new cement make it suitable for applications involving electrical insulation of transducers and lead wires, bonding components to structures subject to mechanical and thermal stress, and thermal barriers requiring rigidity and strength. **127**

Transistor Test Equipment

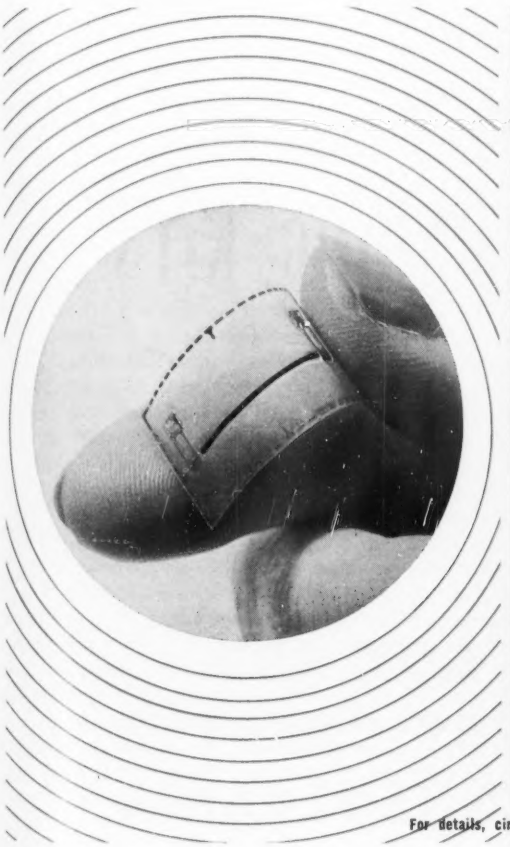
Bulletin 101 describing the F-20 transistor tester is now available from Molecular Electronics, Inc., 85 Weyman Ave., New Rochelle, N. Y., a subsidiary of Precision Circuits, Inc.

The bulletin highlights the principal advantage of the tester, instantaneous indication of gain-bandwidth product and, in addition, lists other features of the F-20 unit. All pertinent specifications and applications also are included. **129**

NAS Research Report

Research Highlights of the National Bureau of Standards, Annual Report 1960, National Bureau of Standards Miscellaneous Publication 237, 189 pp., 65 cents (order from Superintendent of Documents, U. S. Government Printing Office, Washington 25, D.C.).

Research Highlights presents in compact, digest form an illustrated account of National Bureau of Standards research and measurement activities in fiscal year 1960. The Bureau's research program is portrayed, reflecting this agency's role of leadership in this field.



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THE WILLIAM M. MURRAY LECTURE, 1960

The Role of Stress Distribution in Fatigue

Lecturer explains why engineers and designers must be concerned with more than maximum stresses and stress-concentration factors. Also, he considers the fatigue of notched members in terms of crack initiation and crack propagation

by R. E. Peterson

PART I—GENERAL CONSIDERATIONS

Introduction

It is well known to engineers that nearly all machine elements and structural members are of such a shape that the maximum stress cannot be calculated by the elementary basic formulas of mechanics. It is also generally known that only in a few instances can an exact solution be obtained for finite members by even the most sophisticated mathematical analysis. Much of our information concerning practical parts has been obtained by experimental stress analysis (photoelasticity, strain gages, etc.). Nearly all this information is in the form of stress-concentration factors; namely, maximum stress divided by nominal stress (as calculated by the appropriate elementary formula). In very few instances is the complete stress distribution obtained, even though the experimental techniques for doing this are well established. Perhaps the reason for this scarcity of information concerning stress distribution is that, while the designer makes use of the maximum stress (by applying the stress concentration factor), he does not ordinarily make use of the stress distribution and, consequently, investigators tend to be discouraged from spending the time required to obtain this additional information. This is most unfortunate because it is becoming increasingly apparent that to understand the strength behavior of members under service conditions, one must consider the stress distribution as well as the maximum stress.

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Why Stress Distribution at Critical Section Is Needed

Effective Utilization of Surface Hardening. Surface hardening (case hardening, nitriding, etc.) introduces two factors: (*a*) increased strength of hardened region and (*b*) residual stresses. Relative to (*a*), Fig. 1 shows that the fatigue strength of steel specimens can be greatly increased by hardening.[1-3] Specimens for such tests are of uniform hardness so

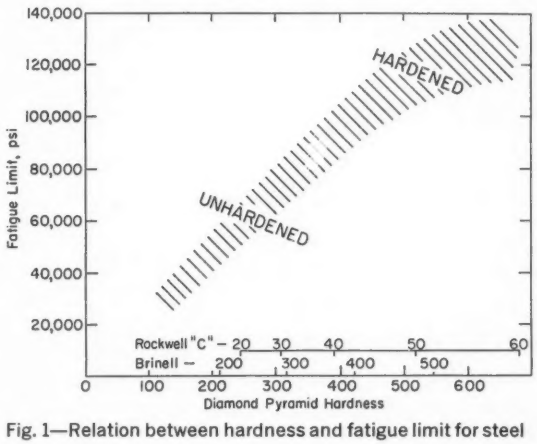

Fig. 1—Relation between hardness and fatigue limit for steel

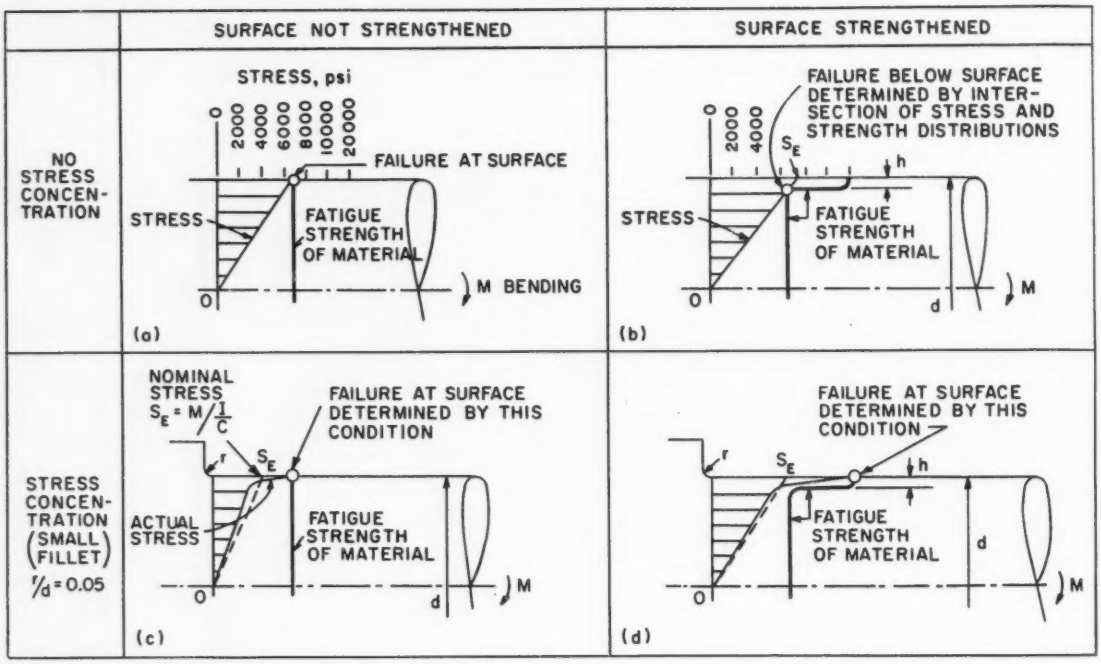

Fig. 2—Conditions of failure for plain and filleted shafts in bending (1-in. diam)

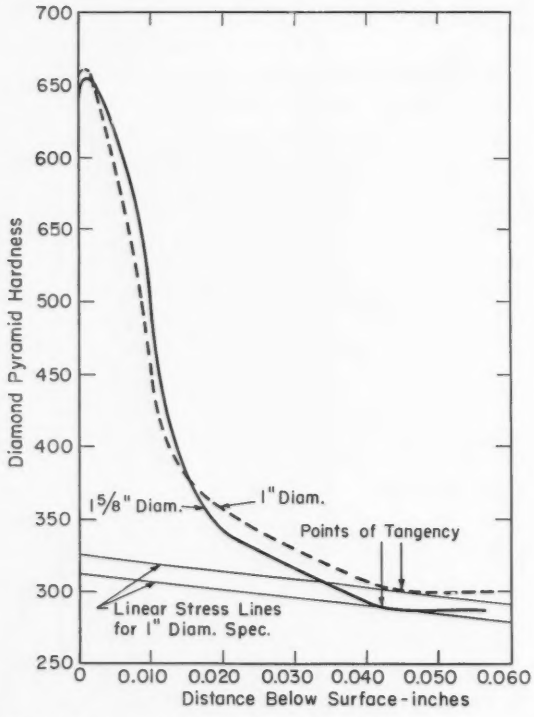

Fig. 3—Hardness distribution curve for nitrided VCM steel (72 hr at 480° C)

that no gross residual-stress pattern is present. Relative to (*b*), residual stresses increase fatigue strength if they are compressive at the critical region, and decrease fatigue strength if they are tensile.[4] These two effects (*a* and *b*) usually co-exist in surface-hardened members, making exact analysis extremely difficult.[4] For the present purpose, that of emphasizing the role of stress distribution, we will consider only the effect of (*a*). If one has detailed information concerning residual stresses and how these are altered by cyclic stressing, one can then complete the present analysis. In Fig. 2 we consider the effect of a surface-strengthened layer on straight and shoulder-filleted shafts in bending. The conclusions will apply also for tension or torsion.

Utilizing a stress scale in the axial direction, we show the stress distribution with a light line and the fatigue-strength distribution with a heavy line. In the absence of hardening, it is assumed for the present that the surface strength is the same as the interior strength. A typical hardness curve for a nitrided layer is shown in Fig. 3. By reference to Fig. 1, the basic fatigue strength due to change of structure *per se* can be obtained, resulting in a distribution not differing greatly from Fig. 3. In each instance we consider that failure occurs when we have increased the loading to where the stress curve intersects or meets the strength curve.[3]

For a straight shaft with a hardened layer, failure should then occur at depth h beneath the surface. That failure does occur beneath the surface in straight shafts has been amply demonstrated[3,5–7] (see Fig. 4), and rough agreement with dimension h has also been found.

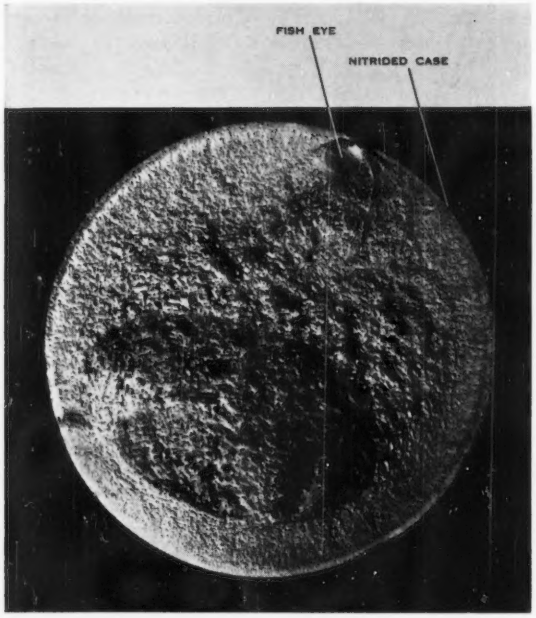

Fig. 4—Fatigue failure. Nitrided bar without stress concentration

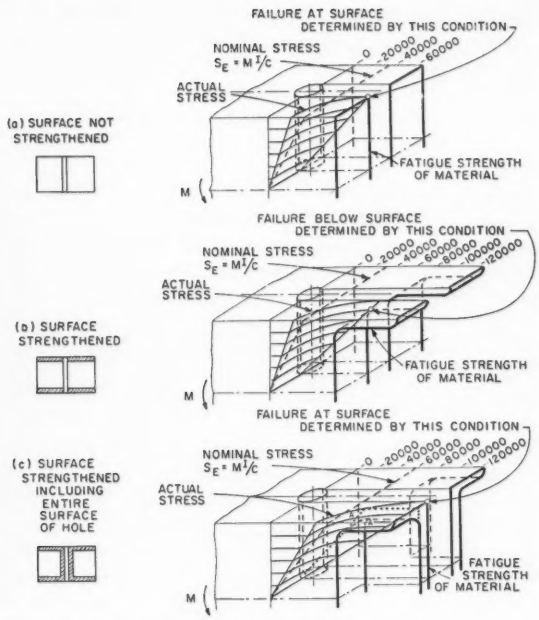
FAILURE AT SURFACE DETERMINED BY THIS CONDITION
NOMINAL STRESS $S_E = M\,I/c$
0 20000 40000 60000
ACTUAL STRESS
(a) SURFACE NOT STRENGTHENED
FATIGUE STRENGTH OF MATERIAL
M
FAILURE BELOW SURFACE DETERMINED BY THIS CONDITION
NOMINAL STRESS $S_E = M\,I/c$
0 20000 40000 60000 80000 100000 120000
ACTUAL STRESS
(b) SURFACE STRENGTHENED
FATIGUE STRENGTH OF MATERIAL
M
FAILURE AT SURFACE DETERMINED BY THIS CONDITION
NOMINAL STRESS $S_E = M\,I/c$
0 20000 40000 60000 80000 100000 120000
ACTUAL STRESS
(c) SURFACE STRENGTHENED INCLUDING ENTIRE SURFACE OF HOLE
FATIGUE STRENGTH OF MATERIAL
M

Fig. 5—Conditions of failure for bending of a bar with a transverse hole

Referring to Fig. 2, for a straight shaft in bending or torsion, the ratio of the fatigue strength of the surface-hardened and unhardened conditions comes out to be, from simple geometry:

$$\frac{\sigma'_f}{\sigma_f} = \frac{1}{1 - (2h/d)}$$

where:

σ'_f = fatigue strength of surface-hardened shaft.
σ_f = fatigue strength of unhardened shaft.
h = depth of hardened layer.
d = diameter of shaft.

We see that the gain due to hardening depends on h/d and, if this is small, only a modest gain can be expected for a straight shaft in bending or torsion. For a straight shaft of 1 in. diam and a layer 0.04 in. deep, the calculated gain is about 9% for bending and torsion. This is in reasonable agreement with the test data.[3] For tension, the calculated gain is zero, for the assumptions utilized. For axial tests, Sutton[7] found endurance limits of 51,000 psi nonnitrided and 52,500 psi nitrided which, considering scatter, seems to be in good agreement with the foregoing.

However, if instead of a straight shaft we consider a member with a shoulder fillet, or other shape resulting in a local stress concentration, we see from Fig. 2 that a large gain in fatigue strength is possible if the stress peak is "contained" in the hardened layer. By judicious use of surface hardening, it has been shown that it is possible to more than double the fatigue strength of a filleted shaft, the gain depending on the geometry of the member.[3] A rather interesting application of the foregoing method of estimating the effect of surface hardening occurred in considering what might be expected for different hardening procedures applied to a shaft with a transverse hole.[3] The analysis is illustrated in Fig. 5 for the bending of a rectangular bar and may be regarded as applicable for a round shaft if the hole is relatively small compared to the shaft diameter. The analysis showed that, for fairly large shafts, a hardened layer of constant thickness on the cylindrical surface only (for example, induction-hardened Fig. 5*b*) has but a limited strengthening effect as compared to a process which also results in hardening the walls of the entire transverse hole (for example, nitriding Fig. 5c). In Fig. 5b, failure occurs beneath the surface, whereas in Fig. 5c failure occurs at the surface, with the stress peak contained in the layer adjacent to the hole surface (dotted construction lines in Fig. 5c show that failure does not occur under the surface, see A-A). Crankshafts which contain transverse oil holes at the journals have been about doubled in

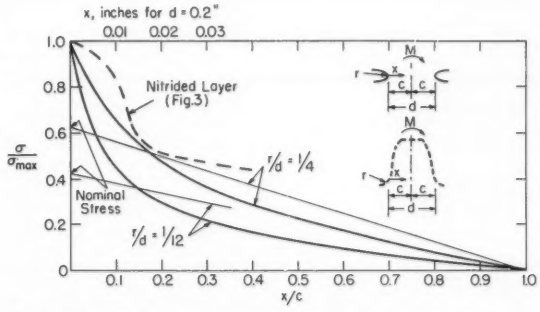

Fig. 6—Stress-distribution curves for deep hyperbolic notched member in bending, with application to 0.2-in. wide gear tooth

fatigue strength by utilizing the above principles.

Decisions concerning the depth of surface hardening of gear teeth cannot be made rationally without consideration of stress distribution. If we consider first the bending of a gear tooth (Fig. 6), then we wish to know the distribution of stress at the fillet at the base of the tooth. While this information can be obtained photoelastically, there is very little to be found in the literature, particularly in terms of principal stresses. However, it has been shown that the stress-concentration factor for a notched bar is similar to, and on the safe side designwise as compared to, a corresponding bar with a shoulder fillet.[8] It seems reasonable, therefore, to use the curves of Fig. 6 for deep hyperbolic notches for application to gear teeth, in the absence of detailed photoelastic data.

Until fairly recently, generated gears had relatively small fillet radii, of the order of $r/d = {}^1/_{12}$, corresponding to the lower curve of Fig. 6, whereas now there is a tendency to go to practically a semicircular fillet with r/d about ${}^1/_4$ (upper curve). To illustrate application of these stress-distribution curves, we show as a dashed line on Fig. 6 the relative strength of the hardness layer from Fig. 3. As can be seen from Fig. 6, both stress peaks are contained within the hardened layer if the knee of the layer is at a depth equal to or greater than about $x/c = 0.15$. As an example we will consider a tooth 0.2 in. wide (about 8 diametral pitch) using the top scale of Fig. 6; a hardened layer of about 0.015 in. is needed to contain both peaks for this tooth size. The "calculated" gain in fatigue strength due to the use of the larger fillet radius is given by the ratio of the nominal stresses, 0.625/0.428 = 1.46, or 46% gain.

As a further illustration, suppose we were to use a case depth half that of Fig. 6, as shown by the top curve of Fig. 7. We see that the stress curve for the smaller fillet is contained within the hardened layer, but that the stress curve for $r/d = {}^1/_4$ reaches the strength curve at A. Failure starts beneath the surface, preventing realization of the full potential strength of the material at the surface. The "calculated" fatigue strength increase due to the larger fillet is 0.482/0.428 = 1.13 or 13%. If we compare this with 46%, we see that the benefit of a hardened layer could be lost if the hardened layer is too thin.

TABLE 1—TYPICAL AUTOMOTIVE CASE-HARDENED PARTS (ASM Metals Handbook[9])

Depth of Case, in.	Typical Part (Low-carbon Steel)
Less than 0.02	Speedometer gears, push-rod balls, shifter forks, tie-rod ball studs
0.02–0.04	Water pump shafts, brake and clutch pedal shafts, valve rocker arms
0.04–0.06	Drive pinions, ring gears, transmission gears, piston pins, steering knuckle pins, roller bearings
More than 0.06	Camshafts

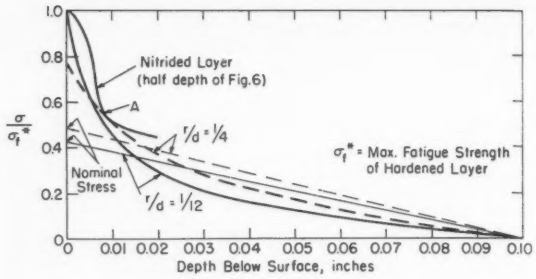

Fig. 7—Stress and strength relations for 0.2-in. wide gear tooth with shallow hardened surface layer

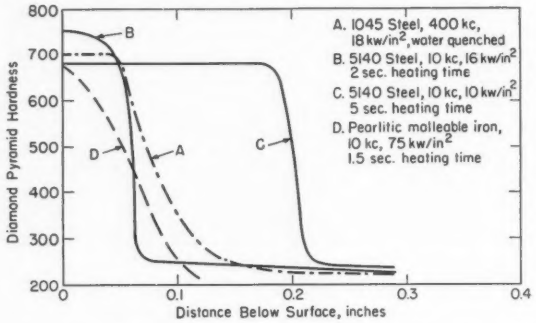
A. 1045 Steel, 400 kc, 18 kw/in², water quenched
B. 5140 Steel, 10 kc, 16 kw/in² 2 sec. heating time
C. 5140 Steel, 10 kc, 10 kw/in² 5 sec. heating time
D. Pearlitic malleable iron, 10 kc, 75 kw/in² 1.5 sec. heating time
Diamond Pyramid Hardness
Distance Below Surface, inches

Fig. 8—Depth-hardness curves for some parts surface hardened by induction (Metals Handbook)

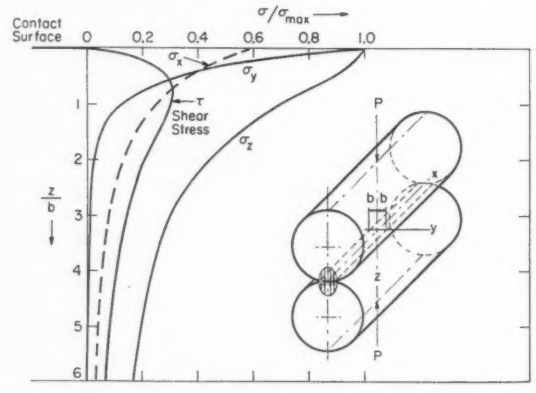

Fig. 9—Distribution of stresses below contact for two axially aligned cylinders pressed together

The practice followed in automotive and other production gearing results in hardened layers with an ample margin of adequacy with respect to thickness (see Table 1). The thickness and shape of the hardened layer can be varied greatly according to the heat-treatment procedure.[9] Some examples are shown in Fig. 8 and while these may not be directly applicable to gearing, they do show the wide variation of case depth obtainable by varying the procedure.

Another consideration in gearing is the effect of surface-contact stresses. The Hertz stress distribution for steel cylinders[10] is shown in Fig. 9 as a func-

tion of b, the half width of the flattened area. As applied to gearing, the width b is calculated as follows:

$$b = 1.52 \sqrt{\frac{P'}{E'} \frac{R_1 R_2}{(R_1 + R_2)}}$$

where

P' = load per unit length.
R_1, (R_2) = radius of curvature of gear (pinion) tooth at pitch line, $R_g \sin \theta$ ($R_p \sin \theta$).
R_g, (R_p) = pitch radius of gear (pinion).
θ = pressure angle.

The hardened layer should be of sufficient thickness to amply contain the stress peaks shown on Fig. 9, taking account of the most severe loading to which the gear teeth are likely to be subjected.*

Other examples of application of specific stress distributions to engineering problems involving surface hardening could be given, but it is believed that the foregoing is sufficient to illustrate this practical type of usage of detailed knowledge of stress distributions.

As mentioned earlier, one can also interpret surface-hardening effects from the viewpoint of residual stresses or, better, from the standpoint of both hardness and residual stress. But, in any event, the distribution of stress (or strain) is vital, and this is the main theme of this presentation.

Analysis of Behavior of Notched Members. There exists a large and rapidly increasing literature on experimental investigations of fatigue-notch effects (grooves, holes, shoulder fillets, keyways, threads, etc.). However, the designer can seldom (if ever) find test data which correspond precisely to the conditions with which he is confronted. Consequently, there is a real need to develop general relations which are in agreement with existing data, and this objective has been pursued by a number of workers over a considerable span of years.[11–16] Correlation methods have been attempted on the basis of stress gradient or volume of material within an arbitrary small percentage of peak stress.[17, 18] These methods, of course, depend on utilization of stress distribution. A more basic solution of the problem is very difficult and will probably not be forthcoming in the near future. It is certain, however, that a complete solution is not possible without knowledge of the stress and strain distribution, not only in the elastic region but also under conditions where the stress peak is in the plastic region. This is an area in which the Society for Experimental Stress Analysis should be expected to make important contributions in the future.

The fatigue process can be viewed as consisting of two phases, crack initiation and crack propagation.[19] Crack initiation results from repeated slip, basically due to shear stress, although the normal stress component probably has some effect. In a material which does not possess a marked directional effect, crack propagation follows the general direction of a principal stress trajectory, although the detailed course is of zig-zag nature in traversing the randomly oriented structure. It turns out that under certain circumstances it is possible to develop crack initiation without subsequent crack propagation. The general

TABLE 2—MATHEMATICAL SOLUTIONS OF STRESS DISTRIBUTIONS

Example	Type of Stressing	Reference
(A) Infinitely wide plate or body of infinite extent:		
1. Circular hole in infinitely wide plate	Tension	Kirsch*
	Bending in plane of hole axis	Goodier and Reissner*
	Shear (by superposition)	Nadai, Baud and Wahl[21]
2. Circular hole near edge of semi-infinite plate	Tension	Mindlin*
3. Multiple holes in infinite plate	Tension	Chih-Bing Ling,* Schoulz,* Hütter*
4. Elliptical hole in infinitely wide plate	Tension	Kolosoff,* Inglis,* Neuber*
	Shear	Neuber*
	Bending in plane of hole axis	Goodier*
5. Semi-circular notch in semi-infinite plate	Tension	Maunsell,[22] Weinel,[23] Weber,[24] Chih-Bing Ling[25]
6. Deep hyperbolic notch in infinitely wide plate	Tension, bending, shear	Neuber*
7. Deep hyperbolic circular groove in infinite body	Tension, bending, shear	Neuber*
8. Spherical cavity in infinite body	Tension	Southwell[26] and Gough
	Shear	Larmar[27]
9. Multiple cavities in infinite body	Tension	Sternberg and Sadowsky,* Atsumi[28]
10. Elliptical cavity in infinite body	Tension	Sadowsky and Sternberg*
(B) Finite Members:		
1. Circular hole at center of plate width	Tension	Howland*
2. Circular hole not at center of plate width	Tension	Sjöström*
3. Semicircular notches in plate	Tension	Chih-Bing Ling[29]
4. Curved bar	Bending	Timoshenko and Goodier[30] (Golovin, Winkler solution)
5. Thick-walled cylinder subjected to pressure		Timoshenko and Goodier[30] (Lamé solution)
6. Rotating disk with central hole		Timoshenko and Goodier[30]

* Detailed reference given in Ref. 8.

* *Since presentation of this paper at the 1960 SESA Annual Meeting, an interesting article dealing with this particular problem has been published in the* SAE Journal, *November 1960, "How to Select Proper Case Depth for Carburized and Hardened Gears," by R. Pedersen and S. L. Rice.*

subject of nonpropagating cracks is receiving considerable attention at present and has an important bearing on the analysis of behavior of notched specimens. In this connection, an attempt has been made at approximate analysis, utilizing the basic crack-propagation data of Frost[20] and the elastic stress distribution for an elliptical notch. Since the details are not likely to be of general interest, the approximate analysis is presented as Part II of this paper. Part II is a self-contained presentation and can be read apart from the general considerations presented in this part of the paper. It was, however, thought appropriate that the paper should include this analysis, since it illustrates another potential usage of stress-distribution information.

Survey of Available Information and Needs

Mathematical solutions are ideal in that they usually cover the complete range of the variables and can be applied in cases where experimental techniques are very difficult or impossible, i.e., extremely small radii. Of the mathematical solutions of interest in the present connection (Table 2), most are for members infinitely wide in two or three dimensions (part A). The stress distribution in a finite member is more difficult to solve mathematically.

Table 2 probably does not represent a complete list, but it is believed that it does cover the main body of available mathematical solutions of interest for design applications. It will be noted that the stress distribution for such an elementary case as an elliptical edge notch is, to the writer's knowledge, not available as an exact solution. For axial stress distribution, the assumption is usually made that the distribution for the elliptical edge notch is the same as for an elliptical hole. In the case of a circular hole in an infinitely wide plate, the stress concentration factor, K_t, is exactly 3, whereas, for a corresponding semicircular edge notch, $K_t = 3.065$. While this indicates that the above assumption for elliptical notches is probably satisfactory for most purposes, it would be desirable to have an exact solution for the elliptical notch.

An observation with regard to Table 2 is that there are fewer items than one might expect, especially with respect to finite members. Also, as originally published a mathematical paper is often not directly usable by engineers. Charts of stress distribution, such as those of Leven[31] for the deep hyperbolic-notch solution of Neuber[11] provide the engineer with an effective means of utilizing the solution.

Turning now to experimental stress analysis (photoelasticity, membrane analogy, etc.), one finds only a relatively small number of instances where the complete stress distribution has been obtained. In photoelasticity most of the available examples are due to Frocht, who has developed a number of special cases.[32, 33]

The situation with respect to photoelasticity may be misleading in that so many fringe photographs have been published. However, even if one plots out distributions from these, one then obtains only $\sigma_1 - \sigma_2$. To separate these, one needs to obtain $\sigma_1 + \sigma_2$ from lateral deformations, or one must use certain special methods.[32]

For engineering purposes we are usually interested only in the stress distributions in the interior beneath the point of maximum stress. Very often this is a line of symmetry (as in the case of a notched or grooved member which is symmetrical in both x and y directions) and in this case the slope-equilibrium method[32] offers a rapid method of obtaining the principal stresses. One must, however, have an additional piece of information, namely, the location of the interior "cupic point" which is obtained from the zero isoclinic pattern by measurement either on a photograph or on the ground glass of the camera. It is strongly recommended that, in all future investigations of symmetrical members, this measurement be recorded since very little additional time and expense is involved.*

Sometimes the stress concentration is symmetrical in one direction but not in the other; for example, a shoulder fillet. Here the stress trajectory beneath the maximum stress is a curved line and the above rapid methods are not usable. A set of isoclinics covering the area of interest is needed in this instance. Any investigation of a general nature should include at least the taking of adequate isoclinic photographs. In certain special cases of complex design detail, one may not have authorization to obtain more than the maximum stress. However, as pointed out, many investigations are broader in nature and are not aimed at specific application (general research, thesis projects, etc.). Here the above suggestions relative to obtaining information on stress distribution should be applied.

PART II—AN APPROXIMATE METHOD FOR ANALYSIS OF CONDITIONS FOR CRACK FORMATION AND PROPAGATION IN NOTCHED MEMBERS†

Pursuant to our general theme of application of detailed stress distribution information, we will consider the fatigue of notched members in terms of crack initiation and crack propagation. With our present state of knowledge, this problem cannot be solved in a completely rigorous manner; but it is thought that enough can be stated to establish a few guide lines and, in particular, to emphasize the importance of stress distribution.

Crack Initiation in Notched Specimens

The ideas in this section have been presented before[34] but will be summarized here and also will be compared with recent data.

* *M. Leven suggests that in addition a photograph of a low-angle isoclinic (say 5 deg) be taken so that an independent check of the stress distribution can be obtained by graphical integration.*

† *The presented lecture consisted primarily of Parts I and III, with a brief summary of Part II given in connection with the section on "Analysis of Behavior of Notched Members." On a first reading it is suggested that Part II be omitted.*

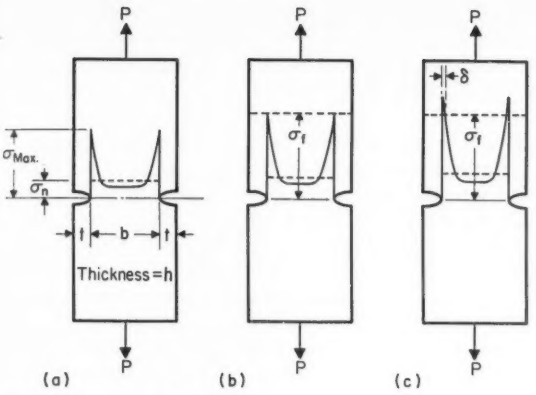

Fig. 10—Condition for initiation of fatigue crack in a notched specimen

Referring to Fig. 10a, let us say that we have a specimen wherein the notch geometry corresponds to a theoretical stress-concentration factor of 4; i.e., for elastic conditions the maximum stress is four times the nominal stress, defined as the load divided by the minimum section area. $K_t = \sigma_{max}/\sigma_n = \sigma_{max}/(P/bh)$. What happens when we make a fatigue test? If the theoretical factor applied, we would expect the fatigue strength of the notched specimen to be $^1/_4$ that of an unnotched specimen. Usually, the fatigue strength is higher than the $^1/_4$ value. If we think about it, this result seems reasonable.

In Fig. 10b let us represent the conventional (unnotched) fatigue strength by σ_f. Let us now imagine that we have applied just enough load to bring the peak stress σ_{max} up to the inherent fatigue strength of the material σ_f as shown in Fig. 10b. Since fatigue initiates on slip planes in crystals (grains), it would be unrealistic to expect that failure would occur under conditions where the maximum stress is applied only at a "point" (or line if we consider thickness). It would seem that it should be necessary to apply more load so that σ_f would be reached at some depth δ as shown in Fig. 10c.

In analyzing fatigue test data, use has frequently been made of the simplest possible assumption concerning δ, namely, that it is a constant for a particular material.[34] An estimated crack-initiation curve for NEL* tests of notched mild-steel† plates[35] as shown in Fig. 11 seems to indicate a variation of δ versus notch radius (Fig. 12), although as seen in Fig. 11 an average value of $\delta = 0.0012$ in. fits the data fairly well. It must also be kept in mind that the experimental points are based on a relatively small number of tests.

At this stage, then, we have a small initial crack of length δ in a notched member. It is known that, under certain conditions of small notch radius, this small initial crack does not propagate.[35]

* *National Engineering Laboratory, East Kilbride, Scotland.*

† *0.22% C, 42,500 yield strength; 63,000 tensile strength; 45% elongation (2 in.)*

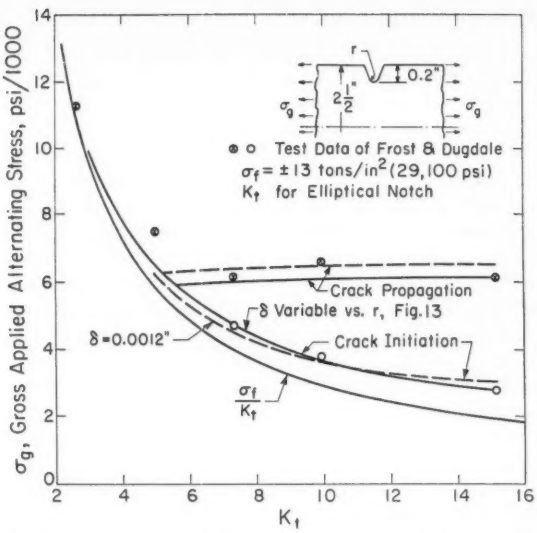

Fig. 11—Fatigue-crack initiation and propagation for a 0.2-in. deep notch in a mild-steel plate

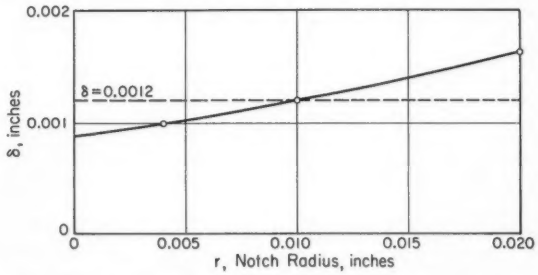

Fig. 12—Values of δ for crack initiation in mild-steel plate

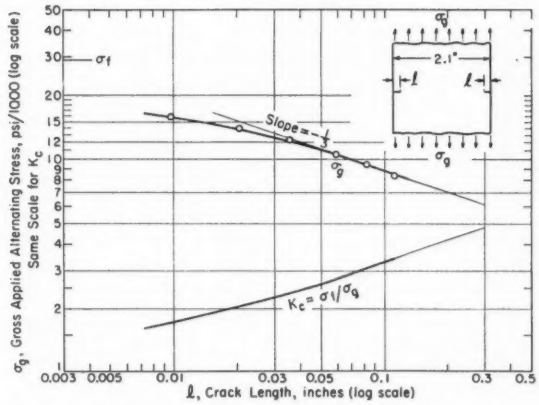

Fig. 13—Alternating stress required to propagate an edge crack in a mild-steel plate (data of Frost and Dugdale)

Crack Propagation

Key information has been developed by Frost[20] in the form of a crack-propagation data for mild-steel† plate subjected to alternating stress (Fig. 13). The crack specimens were prepared by producing small fatigue cracks in a "V-notched" plate specimen (such as shown on Fig. 11), remachining the specimen to remove the edge notches, and finally giving the specimen a stress-relieving heat treatment. In this way, groups of specimens were produced having different crack lengths and, for each crack length, the limiting alternating stress was found which was just sufficient to cause the crack to grow. The light line marked (slope = $-1/3$) in Fig. 13 corresponds to the slope in Frost's paper[20] and is useful for limited extrapolation.

No attempt will be made in this presentation to provide an analytical basis for Fig. 13. Consequently, there will be no discussion of the stress field for thin elliptical notches of zero tip radius, of effect of closing on the compression side of the cycle, etc. We are herein simply accepting Fig. 13 for mild-steel plate and attempting to apply it to notched members.

If we consider the stress fields in the absence of a crack, the only effect of a notch is to introduce a nonuniform stress field in place of the uniform stress field which was applied in obtaining Fig. 13. In Fig. 14 the nonuniform stress distribution is shown as curve OAJ.

Next, the distance δ is laid out (EB) at the fatigue limit stress level σ_f. The crack propagation curve of Fig. 13 (curve KL of Fig. 14a) is extended to E as shown by the dashed line. A curve (BDF) is now drawn displaced horizontally by a constant distance δ. The argument for doing this is similar to that of the previous section.‡

We will now apply the foregoing to members having widely different notch radii, and will consider first a rather large notch radius (Fig. 14a) resulting in a stress-distribution curve which is not steeply sloped (curve OAJ for nominal unit stress). Now, we increase the load until this curve rises to where point A coincides with point B (curve GBCH). Under this circumstance, we would expect the crack to propagate since, if we move slightly to the right of B, we see that we need only a stress level of D to propagate the crack, but we actually have a stress in excess of this at C.

Next, we will examine the situation where a much smaller notch radius (Fig. 14b) results in a much steeper stress curve OAJP for unit nominal stress. Again we apply sufficient load to raise this curve to

† *0.22% C, 42,500 y. s.; 63,000 t. s.; 45% elong. (2 in.)*

‡ *Another procedure is to extend the curve KL directly to B. The results do not differ greatly between these procedures, as is apparent from Fig. 15, since δ is of the order of 0.001 inch.*

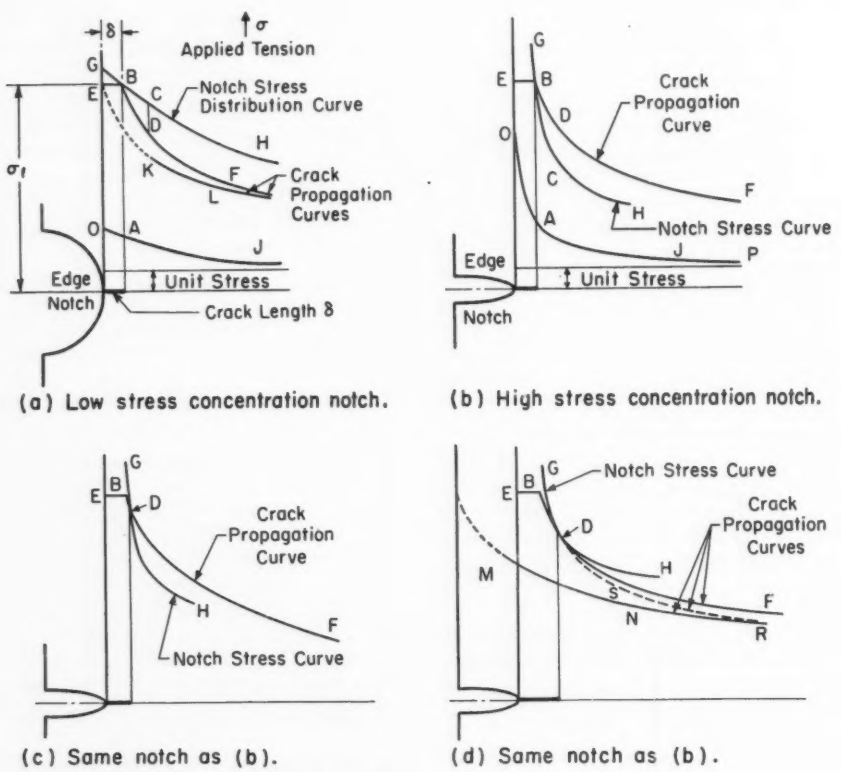

Fig. 14—Method of analysis for fatigue cracks in notched members

where point A coincides with point B (curve GBCH). Here we would not expect crack propagation since, again moving slightly to the right, we see that a stress level D is needed for propagation but we have available only a lesser stress level at C.

A limiting condition, where the crack barely propagates, would be expected to be roughly expressed by coincidence of the stress-distribution curve and the crack-propagation curve.

Returning to the condition pictured in Fig. 14b, let us now increase the load so that the stress curve is raised to the level shown in Fig. 14c. We might now expect that the crack would start to propagate but would cease in the general vicinity of D.

So far we have considered only the region quite close to the notch wherein the nonpropagating cracks are of the order of a few thousandths of an inch in length. It is believed that under these conditions the procedure might provide reasonable results. For longer cracks, however, the method must be modified. Referring to Fig. 14b, we note that the stress peak introduced by the notch has practically vanished at P. Referring now to Fig. 14d and keeping in mind that we are dealing with a relatively wide plate, we would then expect that in the region of P, the crack propagation curve of Fig. 13, starting from the edge of the plate, MNR, would govern, resulting in a crack propagation stress level R instead of F. For the deeper cracks, then, an intermediate curve (dashed curve BSR) would be more realistic. A "correction" of this type was made for the deeper notches; for the shallower notches this was not necessary.

In Fig. 14c, we had increased the loading to where a nonpropagating crack formed. If we further increase the loading to the condition shown in Fig. 14d, a general surplus of stress over the amount needed for propagation is indicated, with the result that the crack in the notched member should "take off" and proceed to complete failure.

Figure 15 shows application of the foregoing method to the estimated fatigue behavior of a $2^1/_2$-in. wide plate, with edge notches 0.2 in. deep and a notch radius of 0.01 in. The stress-distribution curves for various applied stresses σ_g are shown as dashed curves. (σ_g is the gross applied uniform tension stress at an axial distance remote for the notch.) The distribution assumed is that of an elliptical notch in an infinitely wide plate as calculated from the Neuber[11] relations for an elliptical hole. For a finite plate having a low ratio of t/w, notch depth to plate width (in this case 0.08), the stress distribution curve for infinite width may be used with sufficient accuracy. However, there may be a slight difference in stress distribution between the elliptical notch and the 55-deg straight-sided notch used by NEL, for the same t/r (Fig. 15).

Figure 16 shows the crack growth pattern as obtained from Fig. 15, the solid curve corresponding to use of the curve BSR of Fig. 14d and the upper dashed curve corresponding to use of curve BDF. The lower dashed line corresponds to the assumption

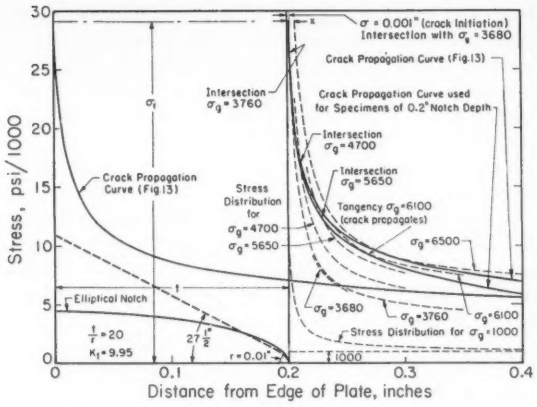
Stress, psi/1000
Distance from Edge of Plate, inches
σ = 0.001" (crack initiation) Intersection with σg = 3680
Crack Propagation Curve (Fig.13)
Crack Propagation Curve used for Specimens of 0.2" Notch Depth
Intersection σg = 3760
Intersection σg = 4700
Intersection σg = 5650
Tangency σg = 6100 (crack propagates)
σg = 6500
Crack Propagation Curve (Fig.13)
Stress Distribution for σg = 4700
σg = 5650
Elliptical Notch
t/r = 20
Kt = 9.95
r = 0.01"
σg = 3680
σg = 3760
σg = 6100
Stress Distribution for σg = 1000
1000

Fig. 15—Stress—strength relations for 0.2-in. deep notch with 0.01-in. notch radius

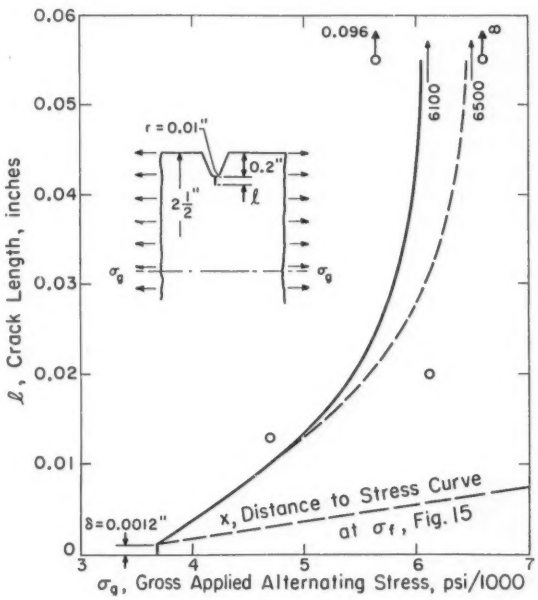

Fig. 16—Crack length vs. alternating stress for notched specimen. (r = 0.01 in.)

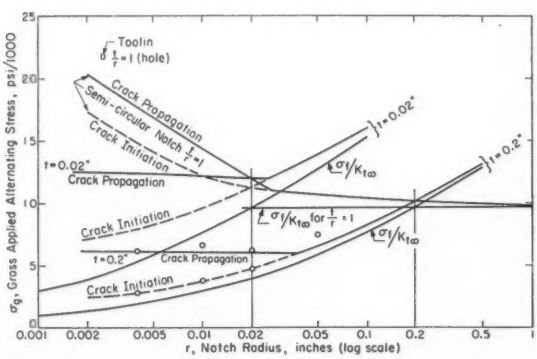

Fig. 17—Fatigue-crack initiation and propagation in mild-steel plate for various notches

that the crack length is given by the horizontal distance to the stress curve at the endurance limit value (x in Fig. 15), but it can be seen that this is not a realistic assumption in that the crack does not "take off." Test points from Frost and Dugdale[2] are shown. These show large scatter as can be expected with small notch radii and conditions close to instability.

Figure 11 shows a summary of results obtained in the foregoing manner for a notch depth of 0.2 in. with various notch radii corresponding to different values of K_t. The upper dashed curve for crack propagation is obtained by use of curve BDF (Fig. 14d) and the upper solid curve by use of BSR. The test points[35] are in rather good agreement with the curves, especially in view of the debatable assumptions on which the method rests.

Similar analyses were made for notch depths of 0.02 and 0.2 in. The results are shown in Fig. 17 as a function of notch radius.* From these curves, one can cross-plot the conditions corresponding to constant notch shapes. In this way curves were developed for semicircular notches, $t/r = 1$, for notches $t/r = 4$ and $t/r = 20$.

Aside from the NEL tests, it is difficult to find comparable data, since not only must the material be comparable but the notch depth must be relatively small compared to the plate width, so that the stress-distribution curves for infinite plate width can be used.

Although not strictly comparable, a test result for $t/r = 1$ is shown on Fig. 17. This value was obtained by P. R. Toolin of Westinghouse Research Laboratories for a 0.3-in. diam round specimen containing a short transverse hole 0.005 in. in diam—the material being 0.15% carbon steel. The test result is reasonably close to the $t/r = 1$ curve.

The results of Fig. 17 can also be put in the form of "notch-fatigue factor" curves (Fig. 18) and "notch sensitivity" curves (Fig. 19) by utilizing the following relations:†

$$K_{fg} = \frac{\sigma_f}{\sigma_{fg}} = \frac{\text{fatigue limit (unnotched)}}{\text{fatigue limit (notched) based on gross stress}}$$

where K_{fg} = notch fatigue factor based on gross stress.

$$q = \frac{K_{fg} - 1}{K_{t\infty} - 1}$$

where $K_{t\infty}$ = stress concentration factor for infinitely wide plate.

It can be seen from Figs. 18 and 19 that for conditions of complete failure the relationship K_{fg} or q versus r, where r = notch radius, consist of two intersecting curves. A number of continuous relationships have been proposed in the past. Perhaps the simplest one is

$$q = \frac{1}{1 + (a/r)}$$

For low-carbon steel of 63,000 psi tensile strength, a value of $a = 0.012$ is found from Fig. 18 of Reference 31; this value has been used to obtain the heavy curve of Fig. 19. It is interesting that this gives a rather good average interpretation of the intersecting curves and in general would be sufficiently good for design purposes.

It was thought at one time that a physical significance could be attached to "a" but the work of Frost and others has made it clear that we are actually dealing with two domains, each being characterized by a different behavior.

How adequate these methods are can only be answered after more test data become available.

It will also be appreciated that, for application to another material, the basic crack-propagation curve as determined from unnotched pieces will need to be available.

Also basic crack-propagation information is needed for round unnotched specimens as well as for flat specimens. In the meantime, one might assume that the curve of Fig. 13 can be used for round specimens of mild steel. An analysis of round specimens carried out on this basis seemed to give reasonable results as compared to test data; however, it was necessary to make several additional assumptions concerning the stress distribution, since the infinite solution was not directly applicable.

The method presented herein should be regarded as a rough approximation which may be useful to engineers who have problems to solve which cannot await a more accurate solution. For a more rigorous solution the following is needed: (*a*) rational interpretation of a fatigue-crack behavior in an unnotched plate in terms of stress or strain, (*b*) solution of interaction of stress (strain) fields for notch and crack.

* *To avoid crowding the diagram, the values for 0.1 in. notch depth, for $t/r = 4$ and $t/r = 20$ are not shown. These results are used, however, in Figs. 18 and 19.*

† *These relations are different from those ordinarily used, but are comparable with sufficient accuracy where the ratio of the notch depth to the plate width is small, in this case 0.08 or less.*

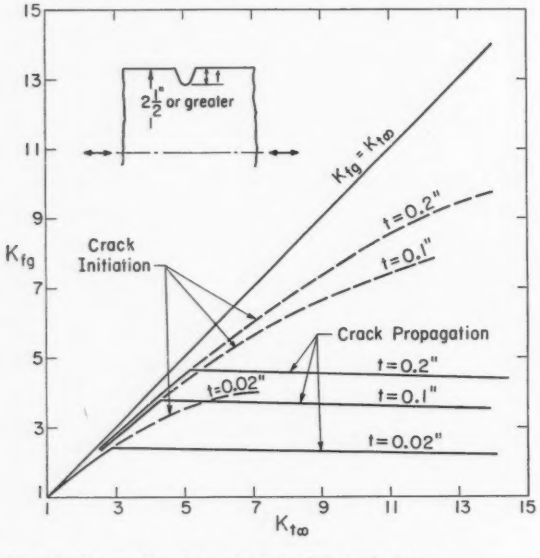

Fig. 18—K_{fg} vs. $K_{t\infty}$ for notched mild-steel plates

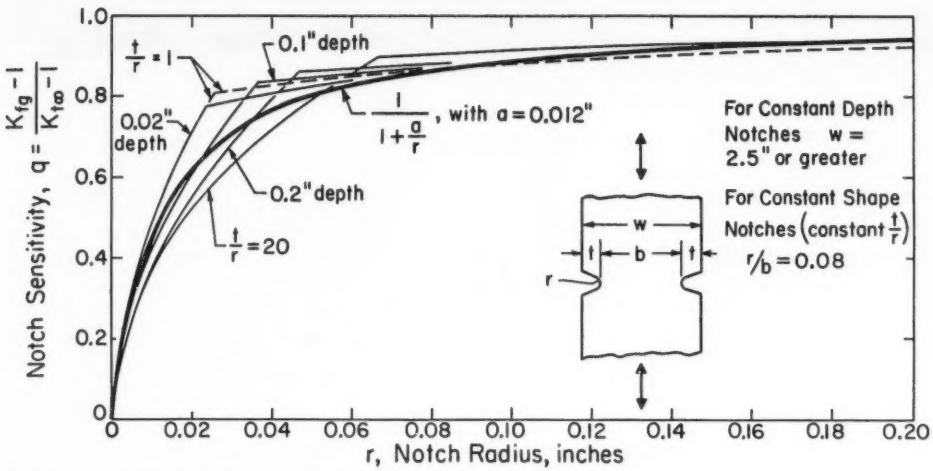

Fig. 19—Notch-sensitivity factor, q, for mild-steel plates

PART III—SUMMARY

It is hoped that it has been made clear that obtaining stress distributions should not be regarded as an academic pursuit. Two reasons have been developed in some detail to show why engineers and designers must be concerned with more than maximum stresses and stress-concentration factors. We have also seen from a survey of existing knowledge of stress distributions that much remains to be done. In fact, with regard to general distribution information for finite members (which, of course, is what the engineer deals with), we have hardly scratched the surface of the knowledge which can be developed and needs to be developed. Technological advances in computers and experimental techniques have opened the door to possibilities of rapid progress in this field in the future. It is the writer's conviction that SESA, by virtue of its unique scope and the qualifications of its members, is destined to play a major role in developing vital information on stress distributions applicable to engineering problems.

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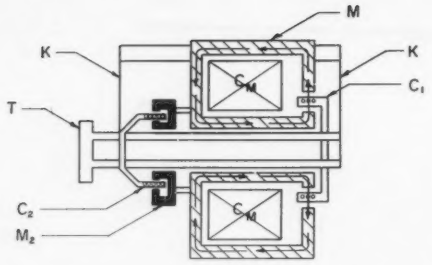

Fig. 1—Schematic cross-sectional view of rectilinear electrodynamic vibration standard

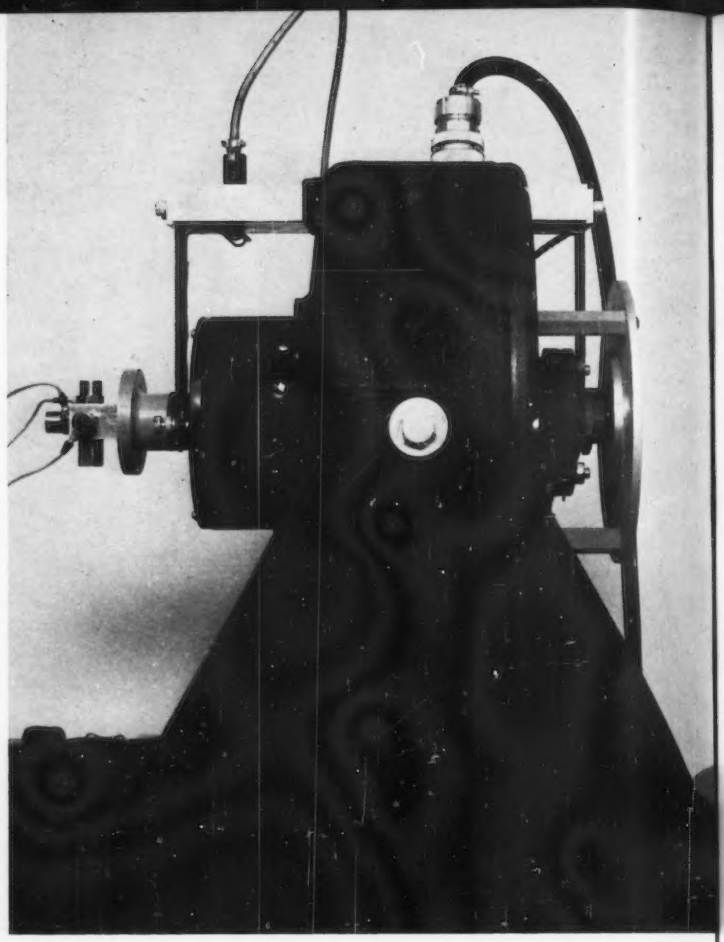

Fig. 2—Test setup for measuring transverse motion of the standard. The three piezoelectric accelerometers attached to the mounting table indicate the acceleration in three mutually perpendicular directions

Improved Standard for the Calibration of Vibration Pickups

Continuing program for developing vibration standards leads to an improved standard free of transverse motion throughout almost all of its frequency range

by R. R. Bouche

ABSTRACT—As part of a continuing program on the development of primary vibration standards, an electrodynamic standard was modified so that it is free of transverse motion throughout almost all of its frequency range. The rectilinear sinusoidal motion produced by the standard is achieved by suspending the moving parts of the standard from the frame by tensioned wires. The standard consists of an electrodynamic vibration exciter equipped with a velocity-sensing coil which is calibrated by the reciprocity method. In the vibration-pickup calibration service recently established at the National Bureau of Standards, the standard is used at frequencies from 10 to 2000 cps at accelerations up to 10 g. The errors of the motion applied to the pickups do not exceed 1% up to 900 cps and 2% between 900 and 2000 cps.

Government and industrial research laboratories use selected electrodynamic velocity, variable resistance and piezoelectric acceleration vibration pickups calibrated on the standard as secondary vibration standards. If properly used these secondary standards can be used to calibrate other pickups by measuring the ratio of their outputs.

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Introduction

During the past decade vibration measurements have become a very important part of the development of space vehicles, missiles, aircraft, ships, machines, etc. There is a continuing demand for improving the accuracy of vibration measurements. The basis for accurate measurements is the primary vibration standard.

This paper reports on an improved vibration standard that was developed as part of a continuing program for developing vibration standards. The standard serves as a primary standard for the calibration of vibration pickups.

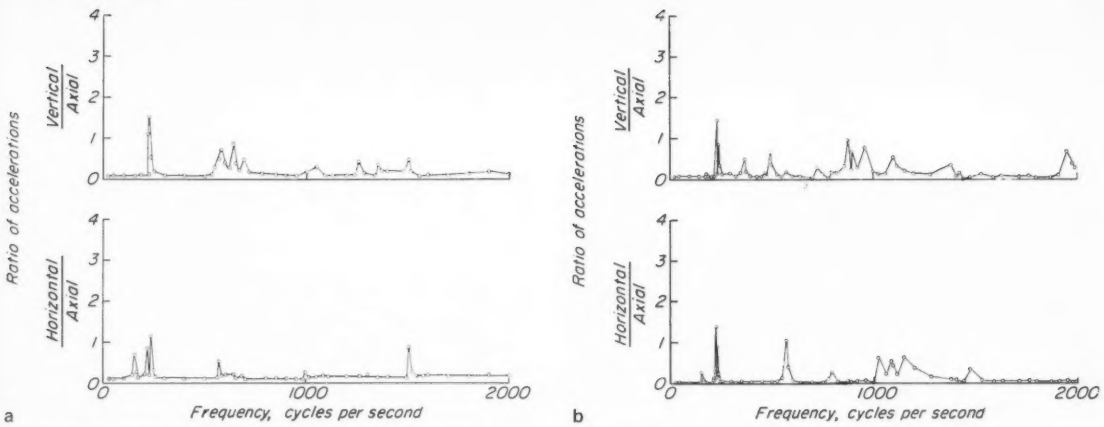

Fig. 3—Horizontal and vertical transverse motion of two unmodified vibration standards

Description of the Electrodynamic Standard

The electrodynamic standard is a rectilinear vibration exciter of the electrodynamic type equipped with a velocity-sensing coil to measure the vibration imposed on the pickup. The standard is illustrated schematically in Fig. 1. The pickup to be calibrated is rigidly attached to table T which, together with the other moving parts, is supported on the main magnet M of the standard by springs K designed to permit motion of the moving parts in an axial direction. The moving parts include the coils and table. The table is set in motion by passing alternating current through the driving coil C_1 which is positioned in an annular air gap in the main magnet, M. The main magnet is excited by passing direct current through the field coil C_M. The resulting axial motion of the table T is measured with the velocity-sensing coil C_2. This is connected concentrically to the table and extends into an annular air gap in the small permanent magnet M_2. The magnet M_2 is flexibly mounted relative to the standard frame in order that the structure and air-gap magnetic field will remain stationary in a fixed reference system, thereby making the generated voltage proportional to the absolute velocity of the table motion.

The standard is mounted on trunnions, allowing the unit to be used with the axial table motion in a vertical as well as a horizontal direction. If the springs K deflect appreciably due to the weight of the moving parts and pickups attached to the mounting table T, provision must be provided to center the coils in their respective magnets. This centering should be achieved in the transverse as well as the axial directions. Centering within $^1/_{32}$ in. is easily achieved although greater off-centering can usually be tolerated before significant changes in the sensitivity of the velocity-sensing coil occur.

The standard is operated, usually, at frequencies above the natural frequency of the single degree of freedom system consisting of the springs K and the moving parts acting as a rigid body. At these frequencies, the acceleration wave form of the table T is similar to the wave form of the driving current in C_1; the wave forms for velocity of table motion and displacement of table motion are smoother than that for acceleration since velocity and displacement are single and double integrals, respectively, of the acceleration. Obviously, it is important that the standard apply pure sinusoidal motion.

The presence of certain resonances in the standard produces motion in other than in the axial direction and thereby distorts the axial motion. Flexural resonances in the shaft which connects the coils, resonances in certain designs of the springs K and other resonances produce transverse motion. Generally the standard is not used at frequencies where there is transverse motion.

Transverse Motion in the Unmodified Standard

The ideal rectilinear vibration standard would produce sinusoidal axial motion at all frequencies throughout its operating range. No transverse or rotational motion would be present. This ideal has not yet been achieved completely; but the amount of transverse motion present at a particular frequency is easily determined.

A simple procedure for measuring the transverse motion in a plane near and parallel to the surface of the mounting table is illustrated in Fig. 2. Three piezoelectric accelerometers with nearly equal sensitivities were mounted on the standard with their principal sensitive axes mutually perpendicular to each other. The response of the accelerometers in planes transverse to the principal axis is approximately 5% of their response along the principal axis. One accelerometer was mounted with its sensitive axis parallel to the axial motion of the standard. Looking at Fig. 2, the axial motion is in a horizontal plane alternately to the right and left across the page. This motion is hereafter referred to as the axial motion or axial acceleration. The second accelerometer has its sensitive axis parallel to the

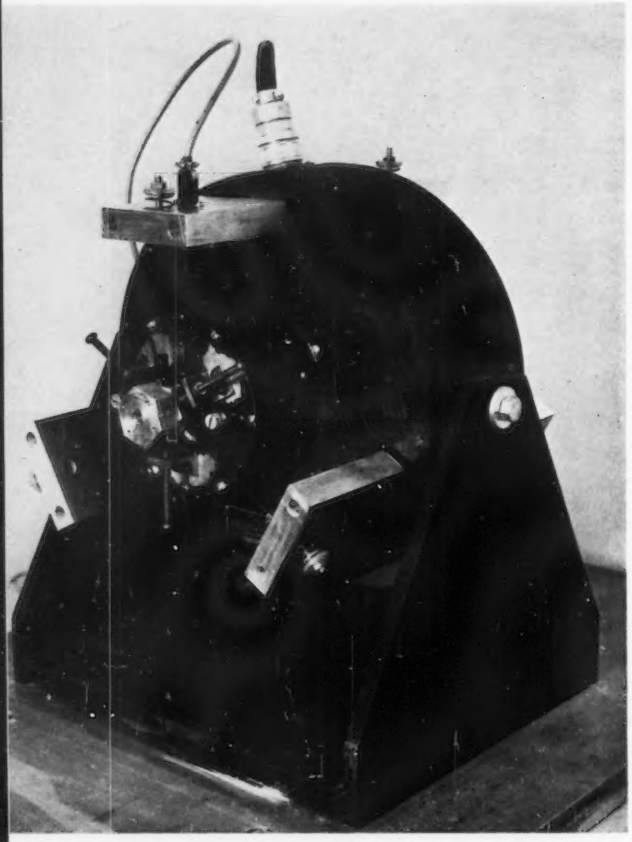

Fig. 4—Vibration standard equipped with a tensioned-wire suspension system which isolates the moving parts from vibration of the frame of the standard

flexure plate springs. It measures the transverse motion in a direction alternately up and down in the plane of the page, Fig. 2. This motion is referred to as the transverse motion in the vertical direction or vertical acceleration. The third accelerometer measures the motion in and out of the page in a horizontal plane, Fig. 2. This motion is referred to as the transverse motion in the horizontal direction or horizontal acceleration. Two dummy accelerometers (one of which is hidden from view in Fig. 2) were attached to balance the load on the mounting table.

Using the above method, the transverse motion in several electrodynamic vibration standards equipped with flexure-plate springs was measured. The results on two standards are shown in Fig. 3. In one standard, Fig. 3 (a), the transverse motion at 230 cps in the vertical direction is 1.5 times the axial motion. Significant transverse motion in the vertical direction is also noted at 580, 640, 690, 1050, 1260, 1350 and 1500 cps. In the horizontal direction, significant transverse motion is noted at 150, 220, 230, 570. 990 and 1500 cps. In the second standard, Fig. 3 (b), there is also considerable transverse motion. The continuous indication of transverse motion of

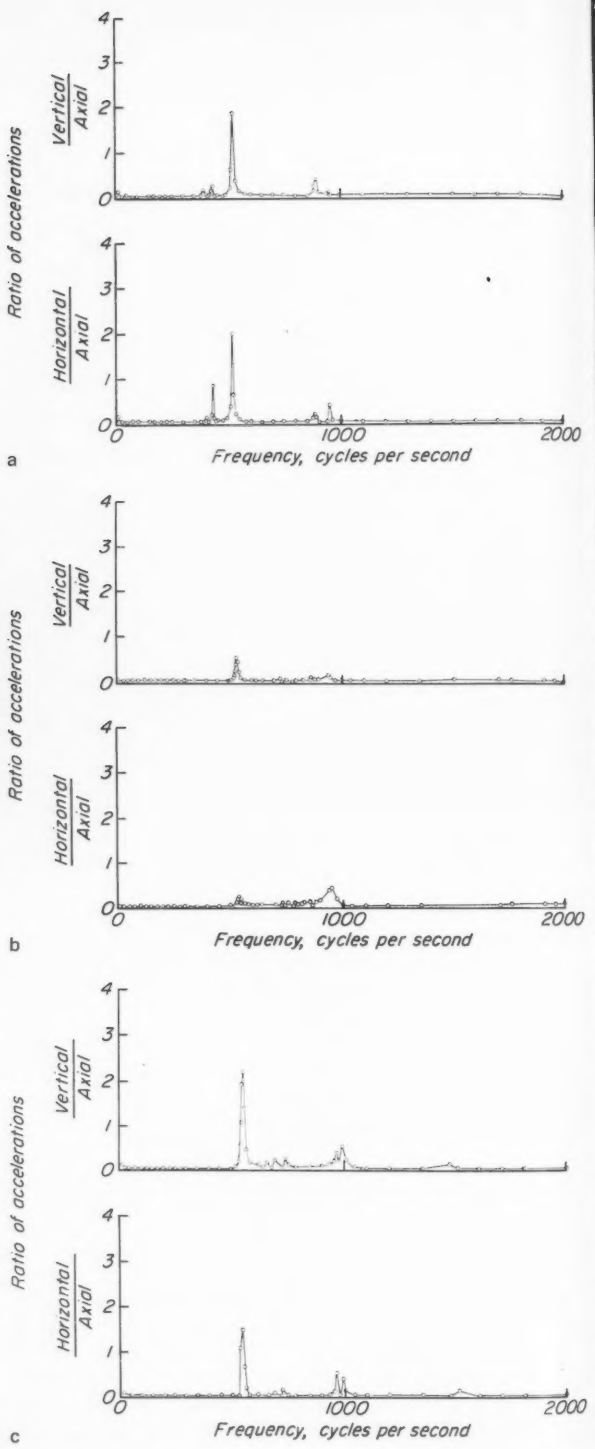

Fig. 5—Horizontal and vertical transverse motion of the standard equipped with tensioned-wire suspension systems: (a) standard equipped with 0.037-in. diam wires under high tension loads, (b) standard equipped with 0.037-in. diam wires under low tension loads, and (c) standard equipped with 0.018-in. diam wires under low tension loads

the order of 0.1 (10%) or less in Figs. 3 and 5 should not be ascribed to transverse motion in the standard since this is not significantly greater than the transverse sensitivity of the accelerometers themselves.

The theoretical bending-mode natural frequencies of the flexure-plate springs and the approximate fundamental bending-mode natural frequency of the shaft and coils were computed. These computations indicated that the transverse motions at 230 cps and near 600 cps were due to resonant response in the flexure-plate springs and shaft, respectively. The transverse motions at a few other frequencies corresponded closely to higher natural-frequency modes of vibration in the flexure-plate springs. However, the transverse motions occurring at many frequencies were not explained by the computations.

It became apparent that much of the transverse motion did not originate in the moving parts or flexure plates. The flexure plates are quite rigid in directions other than the axial direction, therefore motions in the frame of the standard and its mounting block are easily transmitted to the moving parts. In order to eliminate this coupling, the flexure-plate springs in the standard were replaced by a tensioned-wire suspension system that isolates the moving parts from vibration of the frame of the standard in all directions at frequencies above 30 cps.

Tensioned-wire Suspension System

The standard, assembled with the suspension system consisting of six tensioned wires, is illustrated in Fig. 4. One end of each of the three wires, 120 deg apart, is attached to the front portion of the moving parts. The other end of each wire is connected to the frame by passing it through a compression helical coil spring (10 lb per inch spring constant) and terminating it in such a way that the tension load in the wire can be easily adjusted to compensate for the static deflection of the moving parts when adding a pickup to the mounting table. The three other wires connected in a similar way to the back portion of the moving parts are out of view in Fig. 4. The long screws between the wires prevent extreme sideways deflection when attaching and removing pickups. For operation in a vertical plane, an adjustable spring is provided to support the weight of the moving parts and pickup.

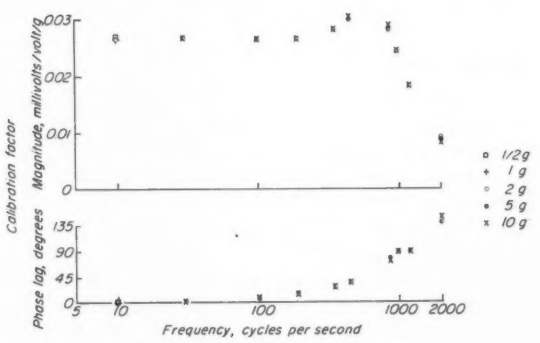

Fig. 6—Calibration factor of a variable-resistance unbonded strain-gage acceleration pickup measured on the standard

Two different sizes of music wire, 0.037 and 0.018 in. in diameter, have been used in the standard with about equal success. The transverse motion was measured with the standard assembled with both sets of wires.

Figure 5 (a) illustrates the transverse motion of the standard assembled with the 0.037-in. diameter wires under high tension loads. That is, the tension loads in the two vertical wires were about 9 lb and that in the four side wires about 8 lb. The fundamental bending-mode resonances of the vertical and side wires appear at 430 and 400 cps, respectively. The second-mode resonances of the vertical wires occur near 890 cps. which is almost exactly twice their fundamental resonance. The second-mode resonance of the side wires and higher-mode resonances of all the wires, which theoretically occur at integral multiples of the fundamental resonance, did not produce any transverse motion in the standard. The transverse motion at 520 cps is the fundamental bending mode of the shaft and other moving parts. Experiments conducted with shafts of different design indicate that the resonance at 950 cps probably is a second bending-mode resonance of the shaft and other moving parts.

Figure 5 (b) illustrates the transverse motion of the standard assembled exactly the same as in the case of Fig. 5 (a), except that the wire tension loads were reduced to about 2 and 1 lb in the vertical and side wires, respectively. The only significant transverse motion in the standard occurs at 530 and 950 cps, the fundamental and second-mode resonances, respectively, of the moving parts. There is no transverse motion due to resonant response in the wires.

Finally, the standard was assembled using the 0.018-in. diameter wires. The tension loads in the vertical wires were about 2 lb and those in the side wires about 1 lb. The transverse motion is illustrated in Fig. 5 (c). The fundamental and second-mode resonances of the moving parts occur at 550 and 990 cps, respectively. These resonances are slightly higher than was the case in Fig. 5 (a) and (b), since a slightly stiffer aluminum shaft was used in the standard in the case of Fig. 5 (c). The small amount of transverse motion which occurs near 700 cps is due to resonant response in the wires.

Examples of Pickups Calibrated on the Standard

The vibration standard equipped with the wire-suspension system is used extensively in the calibration service for vibration pickups recently established at the National Bureau of Standards.[1, 2] Industrial and government laboratories use pickups calibrated at the National Bureau of Standards as secondary laboratory standards for calibrating other vibration pickups. The vibration standard is

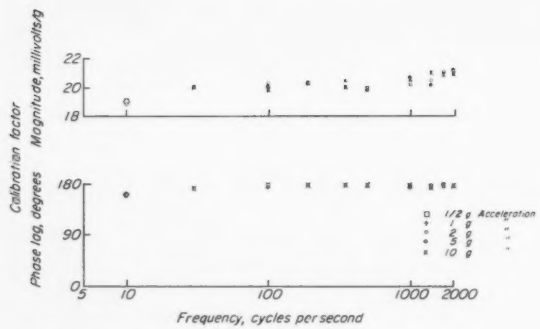

Fig. 7—Calibration factor of a piezoelectric acceleration pickup measured on the standard

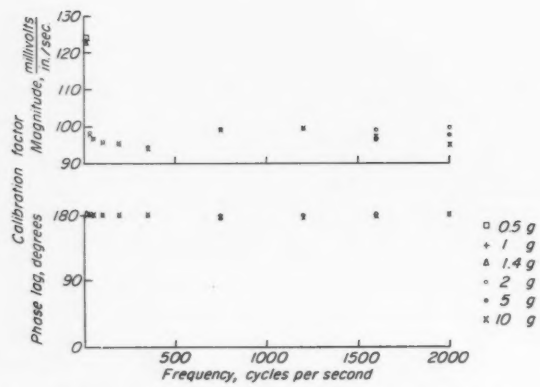

Fig. 8—Calibration factor of an electrodynamic velocity pickup measured on the standard

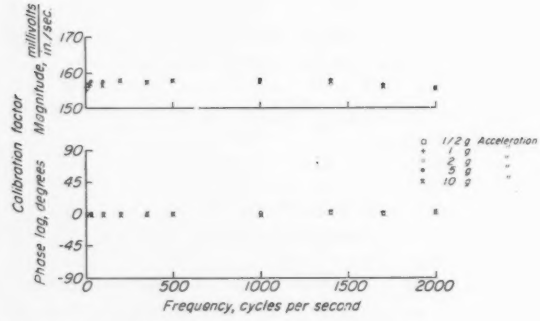

Fig. 9—Calibration factor of a coil and magnet velocity pickup measured on the standard

calibrated yearly by the reciprocity method. Details of the reciprocity method and the methods used for calibrating vibration pickups on the standard are given in References 3 and 4. The estimated errors of the accelerations applied to vibration pickups on the standard do not exceed 1% for frequencies up to 900 cps and 2% from 900 to 2000 cps. Both the magnitude and phase lag[4] of the calibration factor of vibration pickups are determined on the reciprocity-calibrated standard. Phase-lag measurements are useful for determining the undamped natural frequency and frequency range of proportional phase response[4] for the pickup. Examples of pickups calibrated on the standard are given in Figs. 6–9.

The calibration results obtained for a variable-resistance unbonded strain-gage acceleration pickup are shown in Fig. 6. The undamped natural frequency, at which the phase lag is 90 deg, is 1000 cps and the phase lag varies nearly linearly with frequency up to 1000 cps.

Figure 7 illustrates the results obtained for a piezoelectric acceleration pickup. The calibration factor is the output voltage from the cathode-follower preamplifier normally used with the pickup, divided by the acceleration applied to the pickup.

The results obtained for an electrodynamic velocity pickup are shown in Fig. 8. The natural frequency of the pickup is below 10 cps, the lowest frequency at which the calibration was performed.

The calibration results obtained for one type of a coil and magnet assembly electrodynamic velocity pickup is shown in Fig. 9. The calibration factor is constant within one percent. Therefore, this pickup is a more accurate secondary vibration standard than other types of vibration pickups in common use. This pickup is illustrated in Fig. 10. It consists of a small coil (coil plus mounting screw weight is 0.03 lb) and permanent magnet which is suspended by soft flexure-plate springs from an adjustable fixture. The three vertical screws and two micrometer screws in the horizontal plane greatly facilitate centering the coil relative to the magnet in all directions. However, even without these refinements, it is easy to align the coil so that errors due to misalignment are considerably less than 0.5%. The coil and magnet assembly are easily portable.

Discussion

The more general problem of extraneous motions in vibration exciters associated with all types of vibration testing has been discussed by Unholtz.[5] Fortunately, in the case of vibration standards the problem is simplified because, for most vibration pickups, the center of gravity of the load is easily centered on the standard.

The standard equipped with the tensioned-wire suspension system is free of transverse motion throughout almost all of its frequency range. The only frequencies that generally should be avoided are near 550 and 950 cps which correspond to bending-mode resonances in the moving parts of the standard.

Other techniques could be used to reduce or eliminate transverse motion in vibration standards. However, the wire-suspension system proved to be a relatively simple means for reducing considerably the transverse motion in the standard herein described.

In this paper, a simple means for measuring transverse motion is described. To determine completely the motion of a rigid body attached to the standard, it would be necessary to measure the three rectilinear and three rotational motions which correspond to the 6 degrees of freedom. This has been done by Elliott[6] with proximity pickups. However, the simple use of three piezoelectric pickups can detect almost all undesirable motions which normally occur in vibration standards. Furthermore, this latter method is one that can be easily performed in many vibration laboratories.

The axial resonances of the vibration standard have not yet been discussed. The first and second mode axial resonances occur near 2100 and 4500 cps depending upon the mechanical impedance of the pickup attached to the standard. If excessive harmonic distortion is present in the power amplifier driving the standard, these resonances would be excited when performing calibrations at their subharmonic frequencies and some distortion in the axial motion would result. This has not been a serious problem in the standard.

Conclusions

A vibration standard free of transverse motion throughout almost all of its frequency range has been developed. It is used in the recently established NBS service for calibrating vibration pickups at frequencies between 10 cps and 2000 cps at accelerations up to 10 g. The estimated errors of the accelerations applied to the pickups do not exceed 1% at frequencies up to 900 cps and 2% from 900 to 2000 cps.

Velocity and acceleration vibration pickups in common use provide useful secondary standards when calibrated on the vibration standard. A more accurate secondary vibration standard is the small coil and magnet assembly.

Acknowledgments

Lewis C. Ensor and James R. Houghton designed the aligning fixture for the coil and magnet assembly. Ruth M. Woolley performed the calibrations of the pickups. Wilhelmina D. Kroll so kindly reviewed the manuscript and Carol B. Waldron prepared the figures.

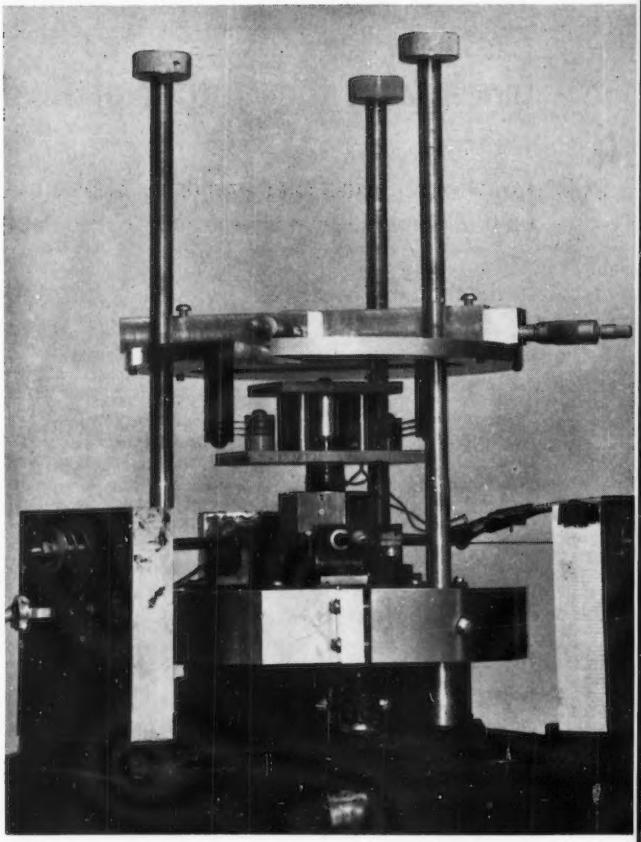

Fig. 10—The coil and magnet velocity pickup of Fig. 9 being calibrated on the vibration standard. The magnet assembly and coil were obtained commercially. The micrometer fixture was added to the magnet to facilitate aligning the magnet relative to the coil

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Linearizing the Output of Resistance Temperature Gages

Investigation indicates that process of balancing Wheatstone bridge and looking up the temperature in a table can be made automatic

by Donald R. Mack

ABSTRACT—The resistance of resistance temperature gages is a nonlinear function of temperature. In addition, the output voltage of a Wheatstone bridge is a nonlinear function of gage resistance. The two nonlinearities oppose each other, and can be made to cancel each other by proper choice of the bridge dummy resistors. The bridge output voltage is then almost exactly proportional to gage temperature change, and can be used to drive any high-impedance indicator or recorder.

Introduction

Resistance temperature gages are small grids of wire, commonly nickel, made to be glued on the surface whose temperature is to be measured. Their resistance, depending on the gage type, is between 50 and 200 ohms at 70° F. A table of resistance vs. temperature is provided for each type by the manufacturer. A typical characteristic plotted from the manufacturers' tables is shown in Fig. 1. This is the characteristic of Baldwin* gage T-7, T-14 and TB-14 and Ruge† types PN-1, PN-2, BN-3 and BN-4.

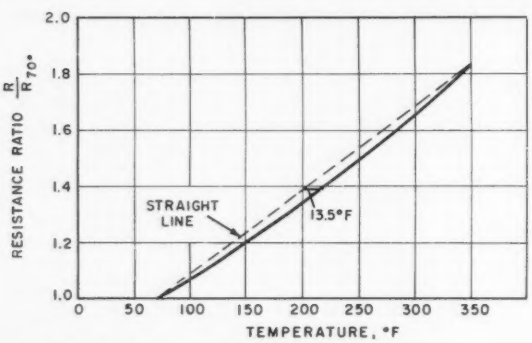

Fig. 1—Typical resistance characteristic of a nickel gage (Baldwin TB-14)

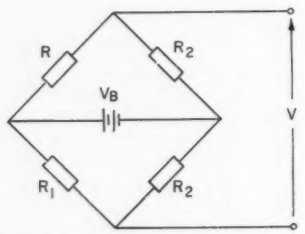

Fig. 2—Simple Wheatstone bridge

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* *Baldwin-Lima-Hamilton Corp., Waltham, Mass.*

† *Arthur C. Ruge Associates, Inc., Cambridge, Mass.*

The resistance is a nonlinear function of temperature. For example, in the range of 70° to 350° F the characteristic deviates 13.5° from a straight line. For this reason, temperatures are usually determined by measuring first the gage resistance on a Wheatstone bridge, and then entering the table to find the gage temperature. This procedure is accurate but tedious. It requires two manual operations: balancing the Wheatstone bridge and looking up the temperature in a table. This paper explains how the process can be made automatic.

Linearizing Circuit

Figure 2 shows a Wheatstone bridge containing a temperature gage R, two equal dummy resistors R_2 and a compensating resistor R_1 which is made equal to the gage resistance at the lowest temperature to be measured. If the output terminals are unloaded, the output voltage V can easily be shown to be

$$V = \left(\frac{R_2}{R + R_2} - \frac{R_2}{R_1 + R_2}\right) V_B \qquad (1)$$

In dimensionless form this becomes

$$\frac{V}{V_B} = \frac{R_2/R_1}{R/R_1 + R_2/R_1} - \frac{R_2/R_1}{1 + (R_2/R_1)} \qquad (2)$$

Figure 3 is a plot of this equation for various values of the resistance ratios over the range of 70 to 350° F. For this range R_1 is $R_{70°}$, the gage resistance at 70° F. The voltage ratio V/V_B is expressed as a fraction of its value at 350° F, the maximum operating temperature of Baldwin gages. These curves show that, for finite values of dummy resistance in the bridge, the output voltage is a nonlinear function of gage resistance.

Fortunately the nonlinearities of Figs. 1 and 3 are in opposite directions, and can be made to cancel each other. Figure 4 shows how they combine for various dummy-resistance ratios. By trial and error, the ratio

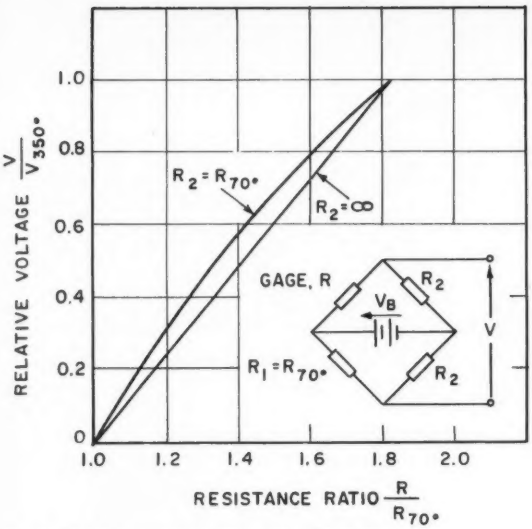


Fig. 3—Relative output voltage of a TB-14 gage

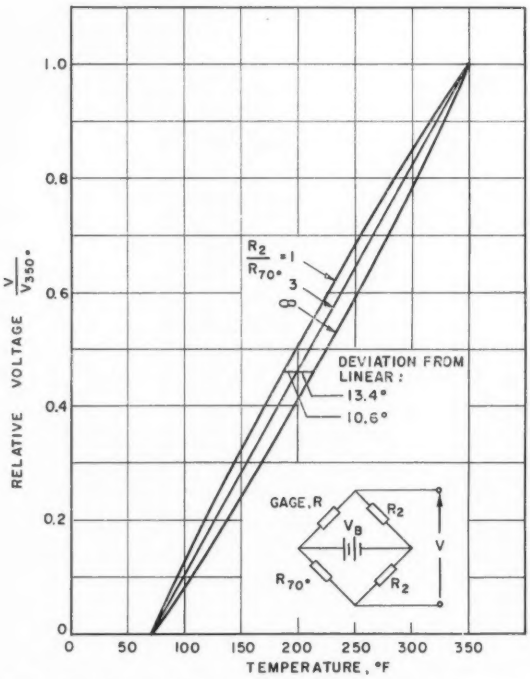


Fig. 4—Bridge output lineared by making $R_2/R_{70°} = 3$

$$\frac{R_2}{R_{70°}} = 3$$

was found to make the bridge output voltage an almost exactly linear function of gage temperature. Samples of the trial-and-error calculation are shown in the Appendix. For the Ruge SN-1, RN-1 and BN-5 gages, the optimum value for the range of 70 to 600° F was found to be

$$\frac{R_2}{R_{70°}} = 4.8$$

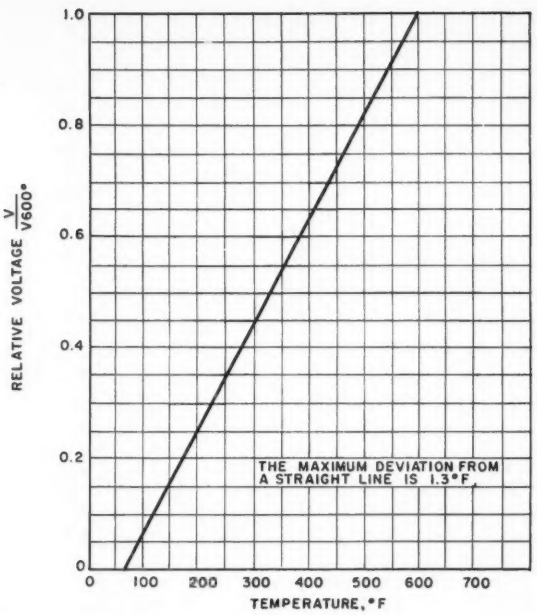


Fig. 5 Ruge SN-1 temperature gage linearized by making $R_2/R_{70°} = 4.8$

Because the calculation of this ratio for any temperature range is a tedious, iterative process, it was programmed for solution on an IBM 704 computer. Subsequently, two more values of the optimum R_2 have been calculated:

Gage	Temperature range	R_2, ohms	$R_2/R_{70°}$	Maximum linearity deviation
SN-1	0–500° F	410	4.1	1.57° F
TB-14	0–100° C	610	3.05	0.19° C

These values of R_2 are optimum to the nearest 10 ohms.

Nonlinearity Error

The deviation of the central curve of Fig. 4 from a straight line is too small to be seen by eye, and must be calculated from the data in the Appendix. The maximum voltage error, found in the last column, is

$$\frac{V}{V_{350°}} = -0.0022$$

Since the slope of Fig. 4 is $1/280, (V/V)/°$ F, the maximum temperature error is

$$T = (280)(-0.0022) = -0.6° \text{ F} \qquad (3)$$

Similarly, the maximum temperature error of the Ruge SN-1 gage in the range of 70 to 600° F is

$$T = (530)(0.0025) = 1.3° \text{ F} \qquad (4)$$

The optimum output-voltage curve of this gage is shown in Fig. 5.

The errors in linearity are approximately the same as those due to error in the manufacturer's published value of $R_{70°}$, which accompanies each package of gages. For most temperature measurements, the deviation of bridge output voltage from a straight line can be ignored.

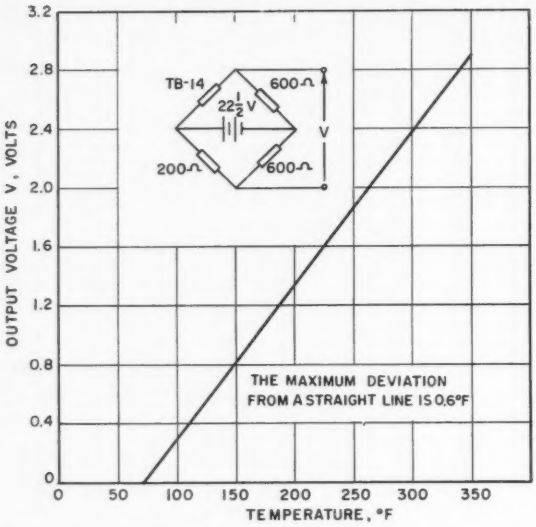


Fig. 6—Output voltage of a linearizing bridge using a Baldwin TB-14 gage

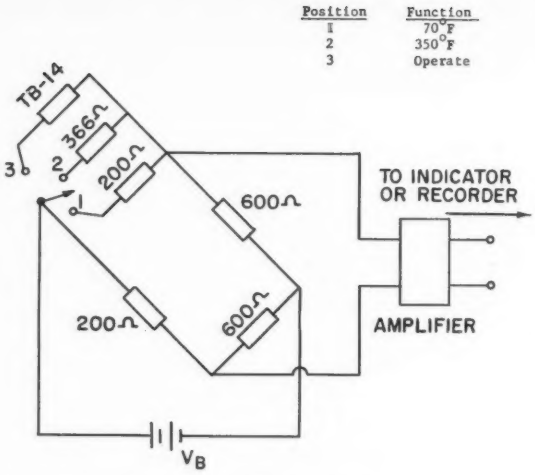


Fig. 7—Linearizing bridge with calibrating resistors

Temperature Calibration

Figure 6 shows a Baldwin TB-14 gage connected in a bridge with optimum dummy resistors for the 70–350° F range. Temperatures can be measured with this circuit by reading the output voltage and multiplying by the proportionality factor. However, the work of multiplication and the error due to the battery drift can be avoided by the addition of calibrating resistors shown in Fig. 7. Positions 1 and 2 of the selector switch give zero and full-scale output of the bridge, and allow adjustment of the indicator to a convenient number of scale divisions.

If some temperature range other than the full range of the gages is to be used, the calibrating resistors can be chosen accordingly. For example, 199 and 298-ohm resistors correspond to temperatures of 20 and 120° C. If the bridge output is to be zero at the lower temperature, the compensating resistor in the adjacent leg will have to be made equal to the lower calibrating resistor.

The resistance of temperature gages at 70° F may vary slightly from the standard 200 ohms (or 50 ohms). The manufacturers publish the exact 70° resistance (within the accuracy tolerance) for each gage. Since the ratio $R/R_{70°}$ at any temperature is very nearly the same for all gages of the same type, the resistance at any temperature can be computed from the published values of $R/R_{70°}$ and $R_{70°}$ for a particular gage. Thus if the 70° F resistance of a TB-14 gage is 201 ohms, the calibrating resistors for 70° and 350° F should be 201 and 368 ohms.

Experimental Verification

The circuit of Fig. 7 was constructed to test the accuracy of this analysis. The TB-14 gage was cemented to a small steel block and immersed in a beaker of oil, together with a mercury thermometer. The bridge output drove a Varian G-10 recorder with 100-division chart paper. Calibrating resistors of 200 and 299.5 ohms were used, to establish the range of 20° to 120° C across the width of the paper. A comparison of the indications of the temperature gage and thermometer is shown in Fig. 8. The linearity of the bridge output voltage is apparent. Any deviation of the experimental points from the straight line can be easily explained as experimental error.

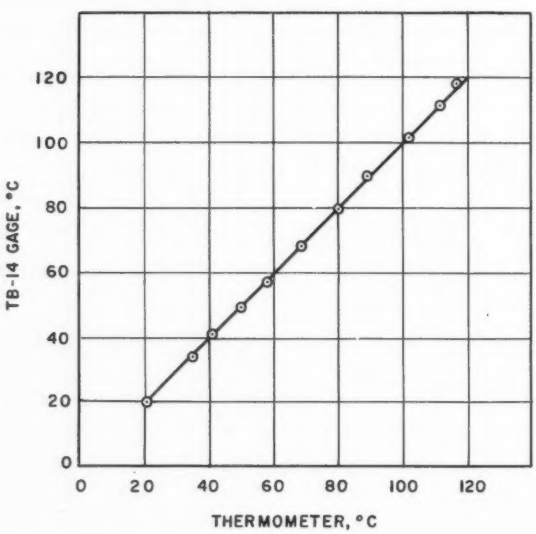


Fig. 8—Comparison of a linearized TB-14 gage with a mercury thermometer

Effect of Finite Bridge Load

The optimum dummy-resistance ratios have been

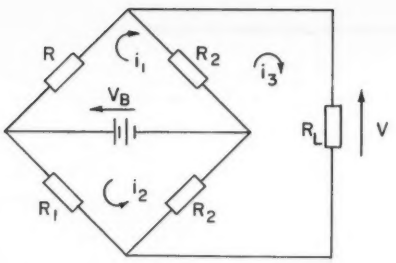


Fig. 9—Loaded Wheatstone bridge

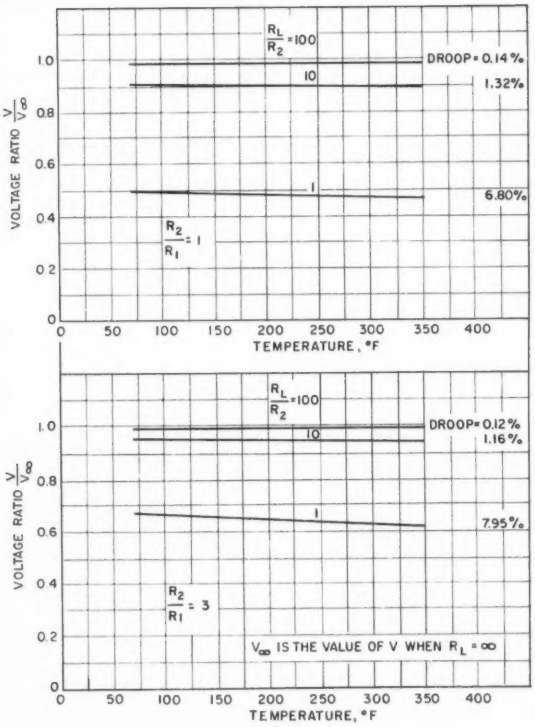


Fig. 10—Effect of finite-load resistance R_L on bridge output voltage V, using TB-14 gages

computed with the help of eq 1, which was derived for a bridge with no output load. Frequently a relatively low-resistance load is desirable. For example, a voltage divider might be inserted in the output to provide a gain control for the indicating instrument. The output load is shown as a single resistance R_L in Fig. 9. The effect of this load on the linearity of the output voltage must be determined.

The three loops provide three voltage equations:

$$i_1R + (i_1 - i_3)R_2 = V_B \tag{5}$$

$$i_2R_1 + (i_2 + i_3)R_2 = V_B \tag{6}$$

$$(i_3 - i_1)R_2 + i_3R_L + (i_3 + i_2)R_2 = 0 \tag{7}$$

They can be solved simultaneously to yield

$$\frac{V}{V_B} = \frac{i_2R_L}{V_B} =$$

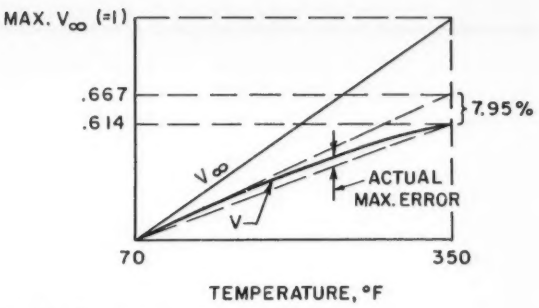


Fig. 11—Nonlinearity due to bridge load

$$\frac{\dfrac{R_2}{R + R_2} - \dfrac{R_2}{R_1 + R_2}}{1 + 2\dfrac{R_2}{R_L} - \dfrac{R_2^2}{R_L(R + R_2)} - \dfrac{R_2^2}{R_L(R_1 + R_2)}} \tag{8}$$

The numerator of this equation is the same as eq 1. The denominator is therefore a measure of the linearity error introduced by the load resistance R_L. Specifically, the deviation from the unloaded output curve caused by the load resistance is the ratio of V/V_B loaded and unloaded. That is,

$$\frac{V/V_B}{(V/V_B)_{R_L=\infty}} = \frac{V}{V_\infty} = \frac{1}{1 + 2\dfrac{R_2}{R_L} - \dfrac{R_2^2}{R_L(R + R_2)} - \dfrac{R_2^2}{R_L(R_1 + R_2)}}$$

$$= \frac{1}{1 + 2\dfrac{R_2}{R_L} - \left(\dfrac{R_2}{R_L}\right)\left(\dfrac{R_2}{R_1}\right)\left(\dfrac{1}{\dfrac{R}{R_1} + \dfrac{R_2}{R_1}} + \dfrac{1}{1 + \dfrac{R_2}{R_1}}\right)} \tag{9}$$

where V_∞ is the value of V when the output load R_L is infinite. This deviation is plotted in Fig. 10 for several values of the resistance ratios.

The meaning of these curves can be explained by a study of the lowest of the six. For this one the optimum dummies R_2 were used, namely three times the 70° F resistance. The load resistance was equal to R_2. For a TB-14 gage the values are $R_1 = 200$ ohms, and $R_2 = R_L = 600$ ohms. The curve shows that at 70° F the output voltage is 0.667 of its open-circuit value. This loss of gain is not serious. However, the fraction decreases nearly linearly with temperature to 0.614 at 350° F. The droop in voltage ratio over the temperature range is

$$\frac{0.667 - 0.614}{0.667} = 7.95\% \tag{10}$$

As Fig. 11 shows, this droop is the departure of the output voltage from the straight line of its original slope. The droop is a pessimistic measure of error. The actual error between the drooping curve and a straight line drawn between its ends is less. However, Fig. 10 shows that, when R_L is increased to 100 R_2 or 60,000 ohms, even the droop is reduced to only 0.12%. The maximum temperature error due to bridge loading is then only

$$(0.0012)(280) = 0.3° \text{ F}$$

The error decreases still further as the load resistance increases.

TABLE 1—SAMPLE CALCULATIONS OF OUTPUT-VOLTAGE LINEARITY FOR THE TB-14 gage

(1) T, ° F	(2) R, Ω	(3) $R + R_2$	(4) $\frac{R}{R+R_2}$	(5) $\frac{V}{V_B}$	(6) $\frac{V}{V350°}$	(7) $\frac{T-70}{280}$	(8) Difference
70	200	800	0.25000	0	0	0	0
100	215	815	0.26380	0.01380	0.1071	0.1071	0
125	228	828	0.27536	0.02536	0.1968	0.1964	0.0004
150	241	841	0.28656	0.03656	0.2837	0.2857	−0.0020
175	255	855	0.29825	0.04825	0.3744	0.3750	−0.0006
200	269	869	0.30955	0.05955	0.4621	0.4643	−0.0022
225	284	884	0.32127	0.07127	0.5530	0.5536	−0.0006
250	299.5	899.5	0.33296	0.08296	0.6437	0.6429	0.0008
275	315	915	0.34426	0.09426	0.7314	0.7321	−0.0007
300	331.5	931.5	0.35588	0.10588	0.8215	0.8214	0.0001
325	348	948	0.36709	0.11709	0.9085	0.9107	−0.0022
350	366	966	0.37888	0.12888	1.0000	1.0000	0

TABLE 2—SAMPLE CALCULATIONS FOR OUTPUT-VOLTAGE LINEARITY FOR THE SN-1 GAGE

(1) T, ° F	(2) $\frac{R}{R_1}$	(3) $\frac{R}{R_1} + 4.8$	(4) $\frac{R/R_1}{R/R_1 + 4.8}$	(5) $\frac{V}{V_B}$	(6) $\frac{V}{V600°}$	(7) $\frac{T-70}{530}$	(8) Difference
70	1	5.800	0.17241	0	0	0	0
100	1.095	5.895	0.18575	0.01334	0.0559	0.0566	−0.0007
150	1.260	6.050	0.20792	0.03551	0.1488	0.1509	−0.0021
200	1.437	6.237	0.23040	0.05799	0.2429	0.2453	−0.0024
250	1.626	6.426	0.25303	0.08062	0.3377	0.3396	−0.0019
300	1.828	6.628	0.27580	0.10339	0.4331	0.4340	−0.0009
350	2.042	6.842	0.29845	0.12604	0.5280	0.5283	−0.0003
400	2.272	7.072	0.32127	0.14886	0.6236	0.6226	0.0010
450	2.517	7.317	0.34399	0.17158	0.7188	0.7170	0.0018
500	2.779	7.579	0.36667	0.19426	0.8138	0.8113	0.0025
550	3.057	7.857	0.38908	0.21667	0.9077	0.9057	0.0020
600	3.351	8.151	0.41112	0.23871	1.0000	1.000	0

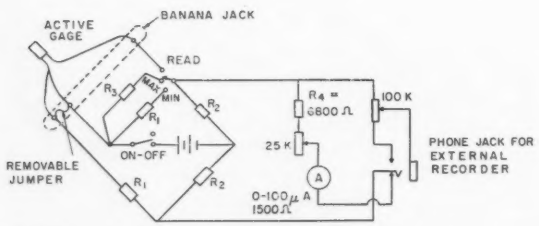


Fig. 12—A practical linearized temperature bridge

A Practical Instrument

Figure 12 is the circuit of a linearized bridge built for factory and laboratory measurements over a variety of temperature ranges. The instrument is shown in Fig. 13. The five R_1, R_2 and R_3 resistors are contained in the plug-in boxes, thus allowing a quick change of temperature range.

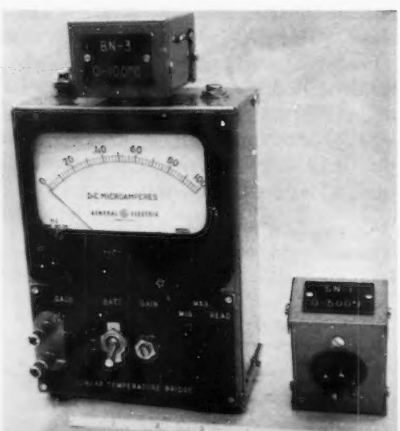


Fig. 13—A linear temperature bridge with plug-ins for various temperature ranges

R_4 keeps the bridge load resistance high enough to prevent significant distortion of the output voltage. The phone jack allows connection of an external high-impedance meter or recorder across a 100 K potentiometer. The jack also disconnects the ammeter, to raise the bridge load resistance.

If the active gage is connected to long, small-diameter leads, changes in lead resistance can be large enough to cause significant error in the temperature measurement. This error can be reduced by connecting three wires to the gage and opening one corner of the bridge. The jumper in the left corner of the bridge in Fig. 12 can be removed for the three-wire connection.

Nomenclature

- R = gage resistance.
- R_1 = gage resistance at the lowest temperature to be measured.
- R_2 = dummy resistance.
- R_3 = gage resistance at the highest temperature to be measured.
- $R_{70°}$ = gage resistance at 70° F.
- R_L = bridge load resistance.
- V = bridge output voltage.
- $V_{70°}$ = bridge output voltage at 70° F.
- V_∞ = bridge output voltage when R_L is infinite.
- V_B = bridge input voltage.

APPENDIX

Calculation of Relative Voltage, $V/V_{350°}$

The first two columns in Table 1 are supplied by the manufacturers for Baldwin T-14 and TB-14,

and Ruge PN-2 and BN-4 gages. The output voltage of an unloaded Wheatstone bridge (see Fig. 2) can be written

$$\frac{V}{V_B} = \frac{R}{R + R_2} - \frac{R_1}{R_1 + R_2} \quad \text{(i)}$$

In the search for a value of R_1/R_2 that will make the output voltage linear, try $R_2/R_1 = 3$. Since for the four gages mentioned above, the 70° F resistance is $R_{70°} = R_1 = 200$ ohms, then $R_2 = 600$ ohms and eq (i) becomes

$$\frac{V}{V_B} = \frac{R}{R + 600} - 0.25 \quad \text{(ii)}$$

Columns 3 through 5 show the calculation of V/V_B. In column 6 each value of V/V_B is expressed as a fraction of the value at 350° F. Column 7 shows the ordinates of the ideal straight-line relation between temperature and output voltage. Column 8 is the deviation of the actual output voltage from the straight line.

For the Ruge SN-1 gage, the best straight line was obtained with $R_2/R_1 = 4.80$. This value makes eq (i) appear as

$$\frac{V}{V_B} = \frac{R/R_1}{R/R_1 + 4.8} - \frac{1}{5.8} \quad \text{(iii)}$$

For this gage the tabular solution and comparison with a straight line are shown in Table 2.

Discussion

by Warren E. Jackson

As the manufacturer of the "temperature gages" which are the subject of this paper, the writer and his associates have appreciated the opportunity of reviewing Mr. Mack's detailed analysis. Mr. Mack has demonstrated analytically that nickel resistance thermometers can be used to provide a high-level voltage/temperature relationship which is linear to within a very small percentage of total span.

The method of linearizing described is particularly applicable when the bridge excitation voltage and output level are not otherwise specified. For an example that would require other linearizing schemes, consider a circuit where 28 v has been specified for excitation. A much larger value of ratio arm resistance than the "optimum" is required to limit the current to a value that will not introduce large self-heating (I^2R) errors. This is particularly true when the fast response and low internal impedance characteristics of the low-resistance thermometer must be maintained.

From our experience, we find that bridge loading is frequently a useful approach in conjunction with the selection of ratio arms to linearize or otherwise shape the resistance/temperature characteristic of nickel thermometers. The loaded bridge is frequently necessary when a relatively low source impedance is desirable.

For a direct deflection indicator such as Mr. Mack has employed, bridge loading is reasonably significant and should be considered in the linearity calculations. Mr. Mack has described this effect in his paragraph entitled "Effect of Finite Bridge Load." In the example cited, it is shown that for a bridge load of 100 R_2 or 60,000 ohms (R_2 being the ratio arm), the temperature error introduced is negligible. This is a rather high total resistance for a direct deflecting indicator. It is perhaps more convenient to use the Thevenin equivalent to investigate the effect of bridge loading than the algebraic expression used by Mr. Mack.

The versatility of plug-in units should make the instrument particularly valuable in laboratory work when a variety of "Stikon"-type resistance thermometers are used for various temperature spans that may be encountered.

Summarizing, Mr. Mack has clearly, accurately and concisely indicated the ease with which resistance thermometers can provide a linear output over wide temperature spans.

Warren E. Jackson is Chief Engineer, Arthur C. Ruge Associates, Cambridge, Mass.

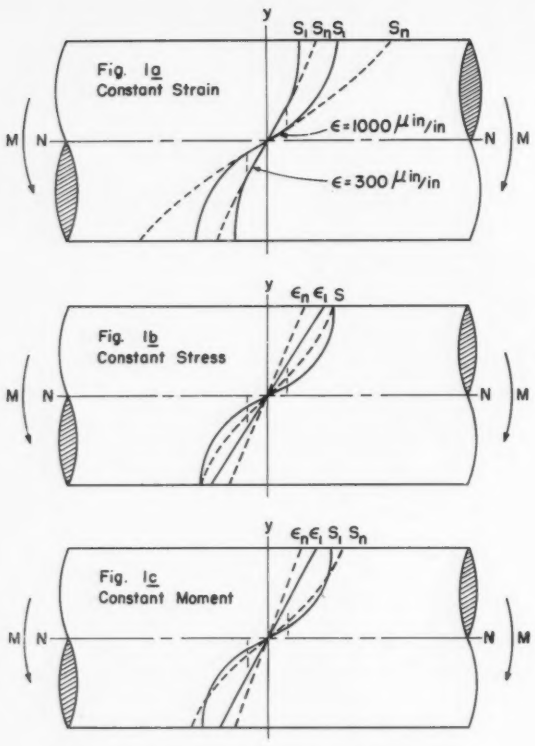
Author's Closure

Mr. Jackson is quite right in pointing out that the microammeter in Figs. 12 and 13 loads the bridge enough to introduce error. This design is justified by two arguments.

1. The R_L/R_2 (Fig. 10) of the microammeter and its series resistors is at least 8300/600 or 13.8. This 8300-ohm bridge load therefore causes a droop (a pessimistic measure of error) of about 1%. Since the microammeter is rated at only ±2% accuracy, the error due to bridge loading is not important.

2. When a high-impedance voltmeter or recorder is plugged into the Fig. 12 circuit to make highly accurate measurements, the microammeter is disconnected from the bridge. The bridge load is then safely high.

Mr. Jackson's technique of using bridge loading to help linearize the bridge, instead of making the load resistance as high as possible, is intriguing. It permits, and in fact requires, leaving the low-resistance microammeter across the bridge when a high-accuracy meter is plugged in. The quick-look temperature indication is therefore always available. An explanation of Mr. Jackson's bridge-loading method of linearizing, with a comparison of the linearities achieved by that method and the method of this paper, would be a welcome contribution.



Note: The solid curves represent the distribution of stress and strain at the first application of the Moment. The dashed curves are the distributions at a later stage.

Fig. 1—Stress and strain distributions in bending. Specimen for three different loading conditions

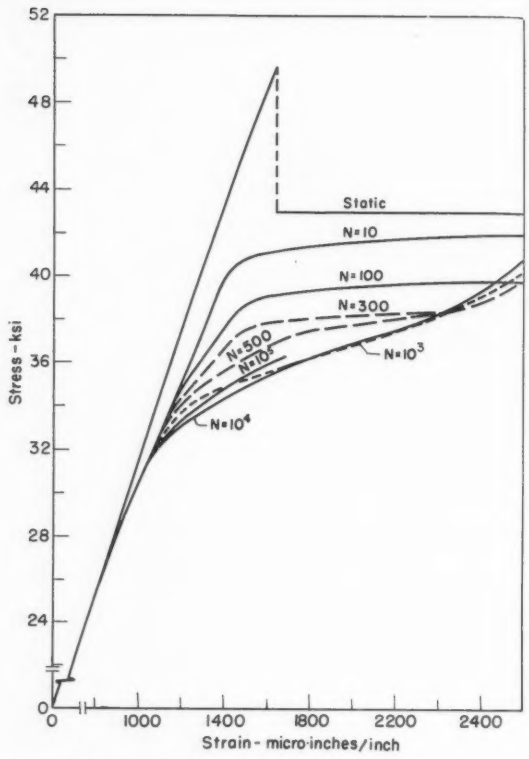


Fig. 2—Cyclic stress-strain locus curves for 1020 steel under axial stress

Stress Redistribution During Bending Fatigue

Relationships among stress, strain and bending moment are examined to determine the distribution of stress in a bending-fatigue specimen and its redistribution during fatigue resulting from change in the cyclic secant modulus of the material

by A. A. Blatherwick

ABSTRACT—One of the effects of the application of cyclic stress, in most metals, is a change in stiffness. Some materials become harder while others initially soften as increasing numbers of cyclic stress are imposed. The effect of this changing stiffness on the distribution of stress during bending fatigue is examined. In general, the maximum stress, strain and bending moment are all observed to vary throughout the fatigue life. The manner of variation is dependent on the type of material as well as the loading conditions imposed. Experimental results are compared with calculated values based on cyclic stress-strain curves obtained in axial-stress tests.

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Introduction

During the last century, volumes of empirical data on fatigue properties of materials have been accumulated. Because of their value to the designer, fatigue data have been obtained on most materials under widely varying conditions. These data are very important and, as new materials are developed, this type of information will continue to be essential.

There are, however, many instances of conflicting reports resulting from differences in testing conditions and procedures. One of the chief differences that has been observed is the variation of fatigue strength with the type of test employed. In general, the fatigue strength determined by axial tests is lower than that found in bending tests. On the other hand, there is no consistent relation between the different types of bending-fatigue tests. In some cases, the constant-deflection type of test gives higher results than the constant-moment type, and in other cases the order is reversed.

A number of investigators have observed changes in certain mechanical properties of materials resulting from cyclic stress.[1–6, 12] Some materials such as mild steel, have been observed to soften under the action of imposed cyclic stress, and their cyclic modulus* decreases during fatigue life. These materials will be described as "cyclic-strain softening." Other materials, such as aluminum and copper, react to cyclic stress in the opposite way and exhibit an increasing modulus with cycles of stress. These will be referred to as "cyclic-strain hardening" materials, or just "cyclic hardening."

These changes in the stress-strain relationship under cyclic stress cause a redistribution of stress during bending-fatigue tests. Because of this redistribution, the results of fatigue tests vary, depending upon the type of test employed.

The purpose of this work is to point out some reasons for inconsistencies in reported fatigue data and to warn investigators of the necessity of making proper adjustments in conducting bending-fatigue tests.

Analysis of Stress, Strain and Bending Moment

Because of the changing cyclic modulus of materials during fatigue, the relationships among stress, strain and bending moment also change. The behavior of these three variables depends also on the external loading imposed on the specimen. Before attempting to make a quantitative analysis of these relationships, it seems desirable to examine them qualitatively.

Stress and Strain Distribution in Bending

Figure 1 is a schematic representation of a longitudinal section of a bending-fatigue specimen showing the distribution of stress and strain. Figure 1a represents the conditions which might be expected for a cyclic-strain-hardening material in which the strain amplitude is maintained constant. In Fig. 1, b and c, the stress amplitude and moment amplitude, respectively, are held constant. In each case, the solid curves are the indicated initial distribution, and the dashed curves give the distribution at a later stage. The strain distribution is assumed to be linear in all cases.

* *The cyclic secant modulus is the ratio of the maximum stress to the maximum strain of the cyclic stress-strain hysteresis loop.*

In Fig. 1a are shown the expected distributions for two levels of strain amplitude (300 and 1000 μ in/in., respectively). In this case, the strain distributions are constant and, therefore, there must be a change in stress distributions in that region of the specimen which is stressed beyond the proportional limit. The greatest change occurs at the outer surface, and the increase is larger for the higher strain condition. Obviously, the bending moment must increase in either of the conditions represented in this figure.

In Fig. 1b, the conditions are given for a specimen subjected to repeated stress in which the stress amplitude at the outer surface is held constant. Because of the hardening effect, it is necessary to reduce the strain as the specimen is subjected to an increasing number of cycles in order that the stress remain constant at the outer surface. This causes a reduction of stress throughout the interior of the specimen. In this case, the bending moment must decrease.

The distributions for constant moment are given in Fig. 1c. In order that the moment remain constant, the increase in stress in the outer portion must be accompanied by a decrease in the inner region. This decrease of stress in the elastic region requires a decrease in strain as shown in the figure. However, the strain does not decrease as much in this case as it does in the constant-stress condition. It is evident, then, that the constant-moment condition is intermediate between the test extremes of constant strain amplitude and constant stress amplitude.

Cyclic Stress-strain Locus Curves

For a series of stress-strain hysteresis loops obtained under axial-stress conditions at different levels of stress, and at the same number of cycles, the locus of the maximum points of these loops (superimposed with a common origin) is called the cyclic stress-strain locus curve.† Figure 2 shows the cyclic locus curves obtained in this way for mild steel by Lidstrom and Lazan.[5] Only the tension half of the curves is shown; experiments indicated that the compression half is identical after a few cycles.

The cyclic locus curves for aluminum were obtained by the method of Lidstrom and Lazan,[5] from bending data under constant moment amplitude. In essence, the method consists of assuming the stress to be a logarithmic function of strain of the form:

$$\left.\begin{aligned} S &= E\epsilon \ (\epsilon \leqq \epsilon_0) \\ S &= \frac{a_0}{X} \log_e X + E\epsilon_0 \ (\epsilon \geqq \epsilon_0) \end{aligned}\right\} \qquad (1)$$

in which

$$X = \frac{\epsilon - \epsilon_0}{\epsilon_k - \epsilon_0} + 1$$

ϵ_0 = strain at proportional limit.
ϵ_k = an arbitrary value of strain.
ϵ_m = strain at outer surface.
E = static modulus of elasticity.
a_0 = an arbitrary parameter.

The integral below yields the moment for a particular value of strain (ϵ_m) at the outer surface:

† *For a more precise description of these curves, see Ref. 7.*

$$M(\epsilon_m) = 2\int_0^c Syda \qquad (2)$$

Where y is distance from neutral axis to the element da, and c is the radius of the cross section. The usual assumption of linear strain distribution over the cross section provides the needed relationship between the variable of integration, y, and strain, and hence stress.

This integration, performed numerically, provides an equation from which the parameter a_0 can be evaluated. This stress function is of course valid only for that particular number of cycles to which the moment-strain data pertained. Successive calculations must be made for each desired number of cycles. Curves obtained in this manner will be presented later with the experimental results.

Determination of Moment-strain Relationship

Using the cyclic locus curves as a basis, the moment-strain relationship can be obtained, for a given number of cycles, by use of eq 2. The stress values are found from the pertinent locus curve. A numerical integration process yields the moment values for the selected maximum strain and the corresponding number of cycles.

Sets of points of M and ϵ_m, thus obtained, may then be plotted to form the moment-strain curve for the given number of cycles. Families of such curves, obtained in this way, are shown in Fig. 3 for mild steel. In this figure, the ordinate values are given as a ratio of moment to the cube of the specimen radius and a factor of 1000, so that they are applicable to any size of specimen.

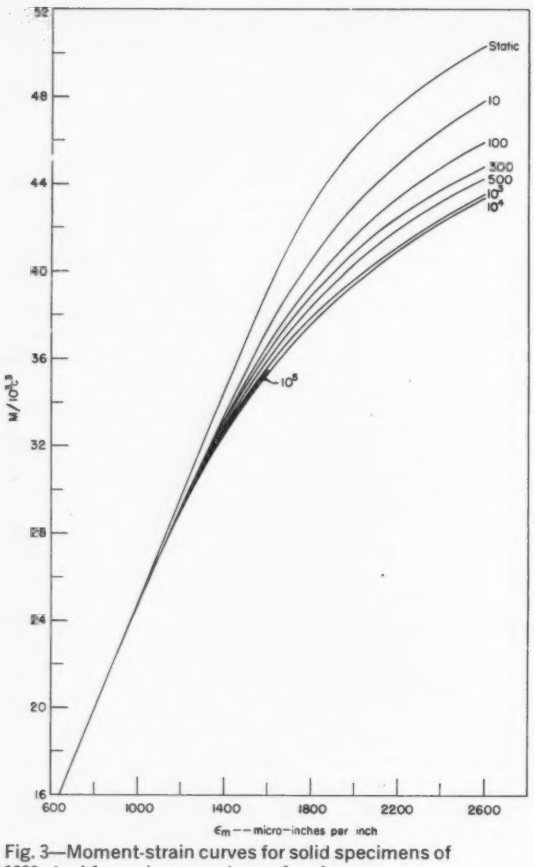

Fig. 3—Moment-strain curves for solid specimens of 1020 steel for various numbers of cycles

Experimental Program

Materials and Specimens

Two materials were chosen for these tests because it was desired to include both a cyclic-strain hardening and a softening type. High-purity aluminum was selected for the hardening type, while SAE 1020 steel was chosen for the softening type. Both of these materials are somewhat extreme in their reactions to imposed cyclic stress in that their stiffness changes considerably (over 100% in the case of aluminum at high strain). They should not, therefore, be considered typical of most structural materials. They were used in this work to emphasize the extreme changes that may occur.

The bending-fatigue specimens were of circular cross section and tapered slightly to provide nearly constant stress over the test length. For mild steel only, hollow cylindrical specimens were also used. Before final machining, the specimens were given heat treatment to relieve stresses and to produce uniform properties throughout. Further details on specimen preparation and heat treatments may be found in references 7, 8 and 9.

Fatigue Testing Machine

The machine used for the bending-fatigue tests was designed and built specifically for this investigation. Two important features were required, (1) ability to vary the stroke at will and (2) provision for determining the bending moment in the specimen and the strain at its outer surface.

In Fig. 4, photographs of the machine are shown. The specimen S is held at the upper end by threaded grips attached to the dynamometer D. The lower end of the specimen is secured by threaded grips to the bending arm A. The lower end of this arm is actuated through the connecting rod by the variable stroke crank C. This crank is so designed that its stroke can be varied while in motion by means of a small motor contained within the shaft housing.

The bending moment in the specimen was measured by means of the dynamometer. Four SR-4 strain gages cemented to the two legs were connected in a bridge circuit so as to be sensitive to bending moment and insensitive to axial forces. The strain was measured by means of a clip gage attached to the two grips at the ends of the specimen. This was done to avoid making any scratches or impressions on the specimen surface which might affect the fatigue life. The clip gage was carefully calibrated against gages attached directly to a specimen identical with those tested.

The outputs of the moment dynamometer and clip

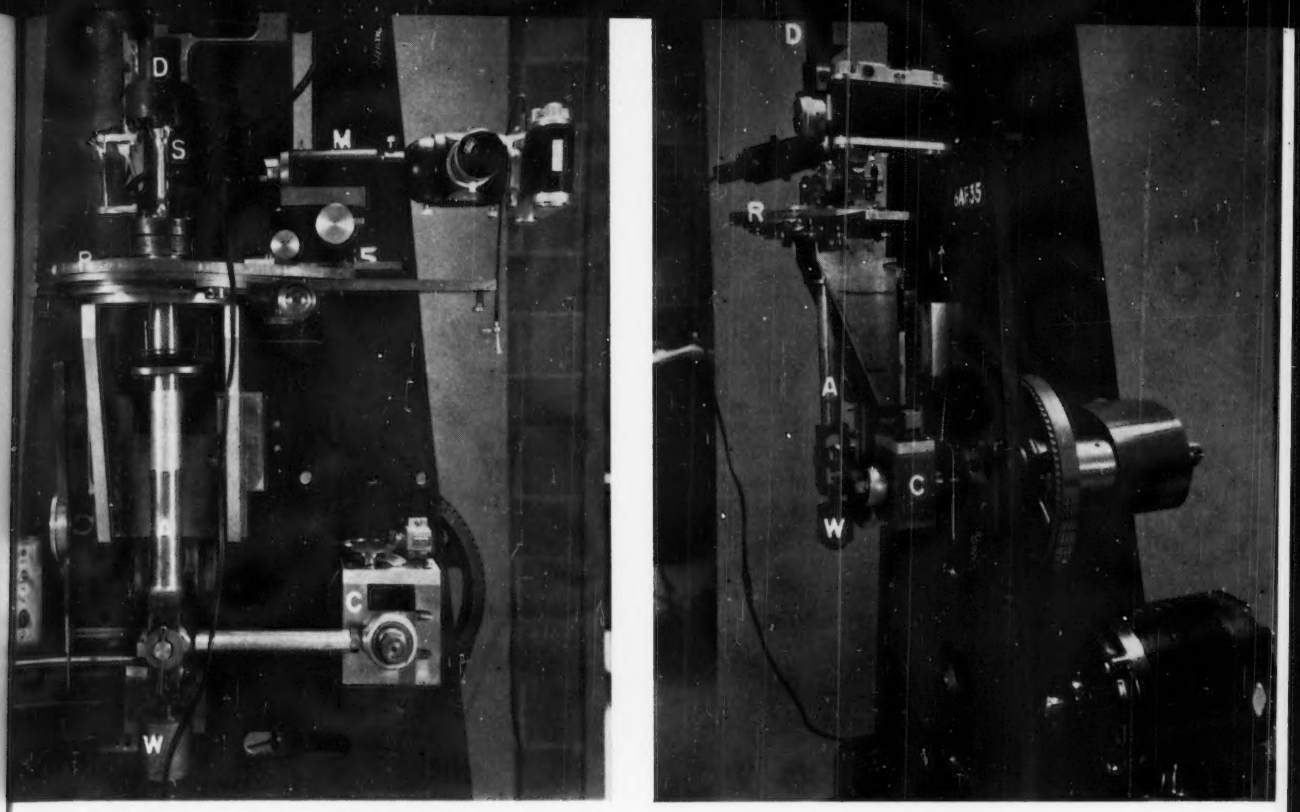

Fig. 4—Photographs of bending-fatigue machine

gage bridges were fed directly to an *x-y* autograph recorder. The moment-strain hysteresis loops were recorded in this manner with excellent repeatability and accuracy to better than $^1/_2\%$ of full-scale readings.

Test Procedure

Bending-fatigue tests were conducted in three different series:

1. Constant-strain tests, in which the strain amplitude at the outer surface of the specimen was maintained constant during the test.

2. Constant-stress tests. In this series, the maximum stress amplitude was held constant throughout the test by varying the strain according to a pattern predetermined from the cyclic stress-strain locus curves.

3. Constant-moment tests wherein the bending-moment amplitude was held constant during the entire test.

At the start of each test, the stroke was set to produce the desired strain (or moment) for the first cycle. Then one cycle was run, turning the crank by hand, and the moment-strain hysteresis loop recorded. For each succeeding cycle, the stroke was adjusted so that the desired level of strain or moment was maintained. After about 10 cycles, the change from cycle to cycle was small enough that it was possible to operate the machine slowly with the motor and to take recordings and adjust the stroke without stopping the machine. Recordings were taken at increasing intervals from this point on.

Loops were recorded at a speed of about 15 rpm, always, but between recordings in later stages of the test the speed was increased to as high as 300 rpm. Tests were conducted using other recording devices, including oscillographs and a Sanborn recorder, to determine the effect of speed on the moment-strain relation. It was found that, within the limits of speeds obtainable with this equipment (0 to 300 rpm), the effect of speed was negligible. Accordingly, between recordings the machine was run at any convenient speed.

Results and Discussion

Cyclic Stress-strain Locus Curves for Aluminum

As outlined previously, the cyclic locus curves for aluminum were developed from the moment-strain data taken during constant-strain tests. In Fig. 5a is a set of moment-strain curves for various numbers of cycles taken directly from these data. These curves are used in other types of tests to predict the pattern of moment variation with strain during fatigue life. In Fig. 5b are shown the cyclic locus curves for various cycles which were obtained by integration of the assumed stress function, described previously. As in the case of mild steel, only the tension half of these curves is shown.

For comparison, the static stress-strain curve for the virgin material is shown as the dashed curve in Fig. 5b. This curve is slightly below the 1 cycle curve, and this is not unexpected. The 1 cycle curve was developed from the moment-strain hysteresis loop which included, of course, the compression half of the cycle. Considerable strain harden-

ing occurs in the first cycle and, therefore, the locus curve should be higher than the static curve.

Constant Strain-amplitude Tests

Figure 6 gives the results of the fatigue tests under constant strain amplitude. In this graph the moment and stress variations are shown by plotting the ratios of the values at a given number of cycles to that at 1 cycle. In this way, the percent change is readily discernible. The moment values were taken directly from experimental data. The stress values, on the other hand, were taken from the cyclic locus curves of Figs. 2 and 5b. The upper curves (above the 1.0 ordinate) pertain to aluminum and the labels on the curves indicate the level of strain amplitude maintained. The curves for mild steel fall below the 1.0 ordinate. In this case scatter bands are plotted, all tests being run at the same level. Also, the results on both solid and hollow specimens are given.

The contrast in behavior of the two materials is evident. The effect of the cyclic hardening of aluminum causes an increase in the moment amplitude of as much as 120% (at the 1000 μ in/in. level) before the development of cracks causes it to drop off sharply. In contrast, the softening in the steel results in a reduction of the moment.

For the steel, calculated moment curves are also shown (double-dashed curves) and they fall within the experimental scatter bands. These curves were calculated from the cyclic locus curves by integrations using eq 2. No calculated curves are shown for the aluminum because they would be identical with the experimental ones. (Recall that the locus curves for aluminum were obtained from these moment curves.)

It is obvious that a bending-fatigue test in which the strain amplitude is held constant would be more severe than a constant-stress test on aluminum. The opposite would be true in the case of mild steel, however.

Constant Stress-amplitude Tests

In these tests, the stress amplitude was held constant by arbitrarily varying the strain according to a pattern determined from the cyclic locus curves. The intersections of a horizontal line at the desired stress level with the different locus curves determined the value of strain that should be imposed at the given number of cycles. The strain patterns which were followed for two different stress levels are shown in Fig. 7. These are the dashed curves. In this case, the mild-steel curves are the upper set while the aluminum curves all fall below the ordinate 1.0. For the aluminum, the strain was held constant for 10 cycles to permit the stress to build up to a fatigue level. It was impracticable to apply sufficient strain at the first cycle to produce the desired stress because of the extremely soft condition of the material. Consequently the aluminum curves are plotted as the ratio of moment or strain at a given number of cycles to that at 10 cycles.

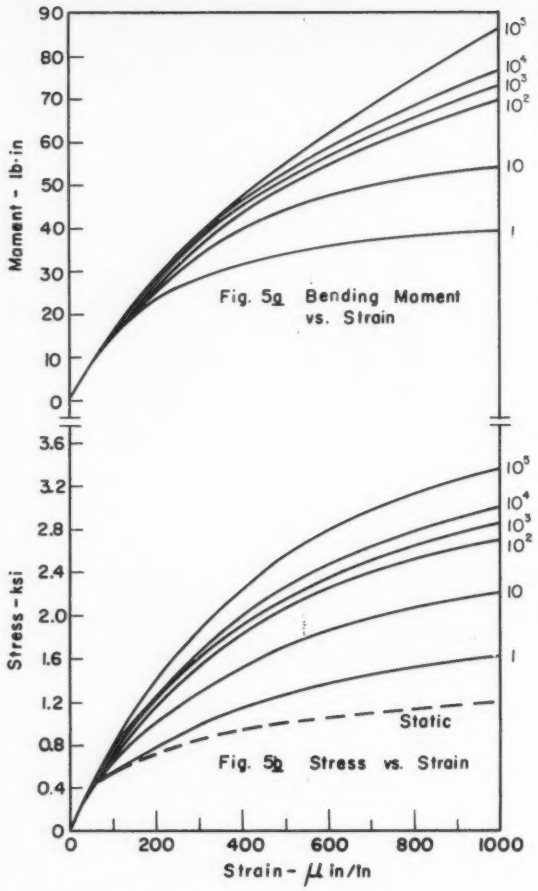

Fig. 5—Bending moment–strain curves and cyclic stress–strain locus curves at various numbers of cycles (aluminum)

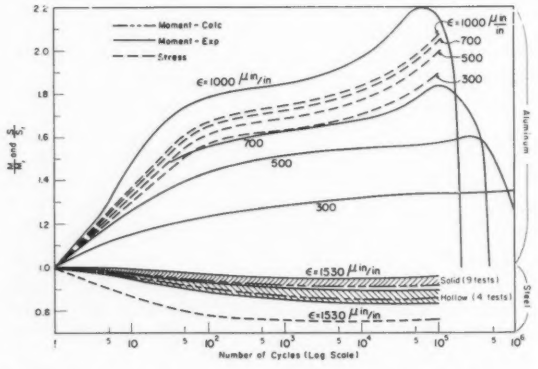

Fig. 6—Variation of moment and stress amplitudes in mild steel and aluminum with cycles of constant strain amplitude

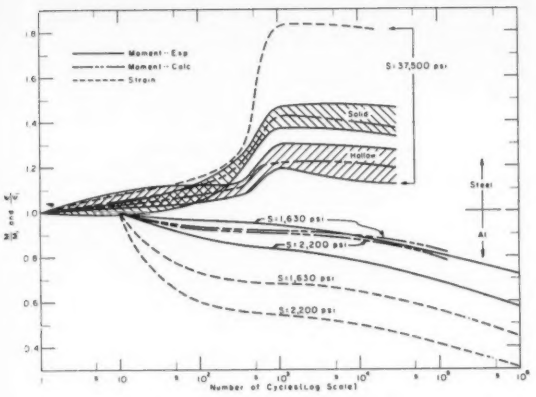

Fig. 7—Variation of moment and strain amplitudes in mild steel and aluminum with cycles of constant stress amplitude

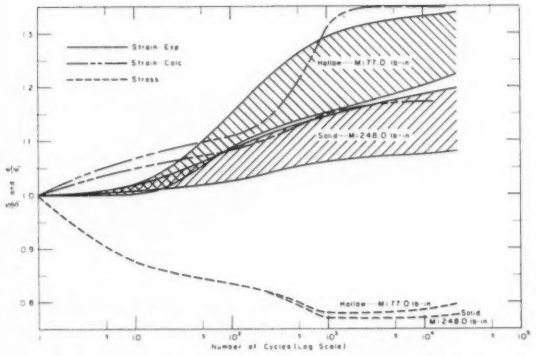

Fig. 8—Variation of stress and strain amplitudes in mild steel with cycles of constant moment amplitude

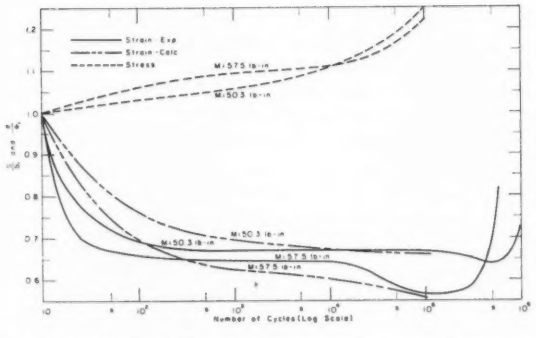

Fig. 9—Variation of stress and strain amplitudes in aluminum with cycles of constant moment amplitude

Here again, the contrast in behavior of the materials is marked. In the case of steel, the strain must be increased considerably (80%) to hold the stress level constant. The moment also increases and the comparison between experimental and calculated results is good. For the aluminum, the strain and moment decrease. The calculated moment curves have the same trend but there is some deviation in actual values. This is the result, at least in part, of using data obtained from constant-strain tests as a basis for computing behavior under constant-stress conditions.

Constant-moment Tests

In this series of tests, the moment amplitude was held constant by adjusting the stroke so that the output of the moment dynamometer was constant. In the case of aluminum, the strain was held constant for the first ten cycles, and then the moment developed at that stage was maintained constant thereafter. This procedure was necessary because the material was so soft initially that an excessive strain would have been required to develop the desired moment during the first few cycles. Because of the relatively few cycles involved in this initial stage, the stress pattern was not significantly affected throughout the bulk of the test.

The results of these tests on mild steel are shown in Fig. 8 and for aluminum in Fig. 9. Again, ratios are plotted, and for aluminum the 10-cycle values are used as a base for the reason given above. The double-dashed curves are the calculated strain patterns for the same moment values. These curves were developed directly from the moment-strain curves of Figs. 3 and 5a.

For the steel, the strain (scatter bands) increases while the stress (dashed) decreases. The calculated strain curves (double-dashed) have the same general trend as the experimental results though differing somewhat in absolute position. Because a greater portion of the hollow specimens is in the region where the change in hardness of the material occurs, these specimens exhibit a greater change in strain than do the solid ones.

The stress in this material decreases as expected. It should be noted again, however, that these are calculated stress values based on the cyclic locus curves. The considerable scatter evidenced by the experimental strain bands indicates that there is a nonnegligible variability in stress-strain properties. This undoubtedly means that if the individual stress values could be determined, there would be scatter bands for stress also.

The results for aluminum (Fig. 9) are reversed from those of steel, as expected. The strain values decrease until cracks become large enough to be felt, then the strain must be increased sharply to maintain constant moment. Good agreement between calculated and experimental curves is evident.

The stresses increase as shown in the figure, but they should actually drop off at the stage where the

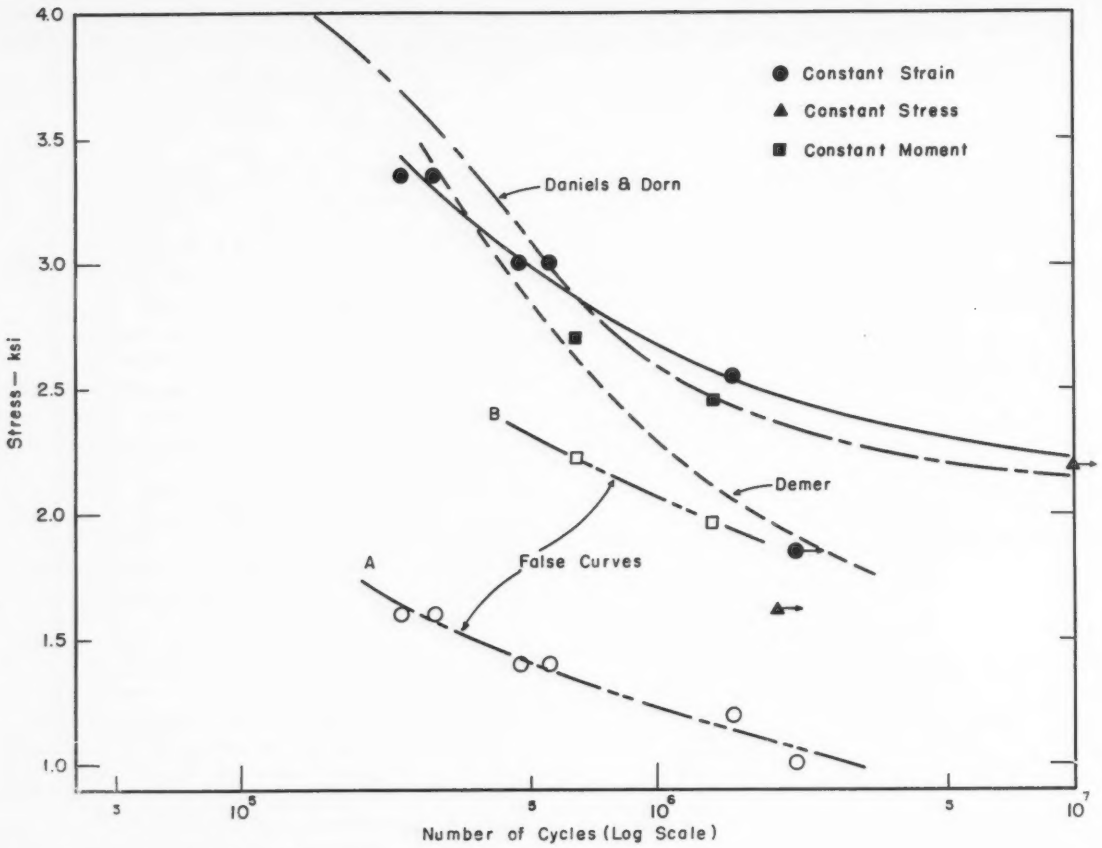

Fig. 10—S–N fatigue diagrams for aluminum

strain begins to increase. Since the cyclic locus curves were not determined beyond 10^5 cycles, it was not possible to calculate the stresses beyond this point.

Fatigue Results

In making comparisons between results of different types of fatigue tests, an immediate difficulty arises as a result of the difference in variable held constant. In the conventional fatigue test, the stress is (or is assumed to be) constant and the usual *S–N* diagram is used to present the data. In the case of constant-strain or constant-moment tests, however, the stress varies throughout the test, as has been noted above. Consequently, there is no common basis for comparison.

A compromise is used to compare the results obtained in these tests in Fig. 10. Here, the results are given for aluminum in the form of an *S–N* diagram, and the values used for plotting points are the stresses at 10^5 cycles in each case. This value was selected because the stress in the specimen did not vary greatly from this level throughout the major part of the test.

On this basis, the solid curve was plotted through the solid circle points which were obtained from constant-strain tests. The triangles are points obtained from constant-stress tests, while the solid squares are from constant-moment tests. Also shown for comparison are initial stress values for each test. The hollow-circle points are from constant-strain data and the hollow squares from constant moment. The two curves, *A* and *B*, which connect corresponding sets of these points are labeled "False" and will be discussed later. Also included in the figure for comparison are the two curves from other sources as indicated.

One of the first points to note is that the constant-strain curve should actually fall slightly lower than its position in this figure, because the stress history during the test was somewhat below the value plotted (stress at 10^5 cycles). Similarly, any curve obtained from constant-moment data should lie slightly below the plotted points shown in this figure (assuming they do represent the fatigue strength correctly). Thus, the true *S–N* diagram should fall somewhat lower than the one shown here. This appears to agree with the data of Daniels and Dorn[10] and of Demer,[11] the latter data having been obtained in this laboratory under axial stress conditions on the same material.

These corrections, however, would be quite small, and it appears reasonable that the actual fatigue strength of this material is represented quite accurately by the solid S–N curve, since the three types of test agree quite closely among themselves as well as with the data from the other sources. The important observation to make, then, is that the "False" curves are considerably in error (100% or more) for this material. These are the curves that would be plotted from the data obtained in the usual fatigue test of this type. Unless adjustments are made during the test, the stress levels in both the constant-deflection and constant-moment tests will increase for this type of material.

To conserve space, S–N curves for mild steel are not shown. However the results were similar to those given above for aluminum. The constant-strain and constant-moment values (when plotted for stress at 10^5 cycles) agree quite well with axial-stress results. However, the "False" curves for mild steel lie *above* the true S–N curve with the constant-moment curve (B) falling between the true curve and the constant-strain curve (A).

Summary and Conclusions

In order to study the distribution of stress in a bending-fatigue specimen and its redistribution during fatigue resulting from change in the cyclic secant modulus of the material, the relationships among stress, strain and bending moment in the specimen were examined. Of the three, strain at the outer surface and the bending moment at a cross section were measured experimentally. From these results the stress distribution was deduced. Comparisons were made between experimental values of bending moment (or strain) at various stages of a fatigue test and values calculated from cyclic stress-strain curves obtained independently. The comparisons were favorable enough to indicate that the assumed stress distributions were reasonably accurate. Fatigue results further substantiated these conclusions.

The following conclusions appear evident from the results obtained in this investigation:

(a) As a result of changing cyclic stress-strain properties, the stress distribution in a specimen changes during bending-fatigue tests.

(b) The nature of the changing stress distribution depends upon the type of material; whether it is cyclic strain hardening or softening. It also depends upon the imposed loading conditions; whether the strain amplitude is held constant, or the moment or the maximum stress. In general, the constant moment-amplitude test is intermediate between the constant strain and stress tests. Whether the constant strain-amplitude test is more or less severe than the constant stress-amplitude test depends on the type of material.

(c) Unless the loading conditions are adjusted frequently, results obtained in bending-fatigue tests may be in error by 100% or more as a result of the stress redistribution. Again, it should be understood that most structural materials do not exhibit the extreme changes evidenced by the materials studied in this work.

Acknowledgment

The author is grateful to Prof. B. J. Lazan, Head of the Department of Aeronautical Engineering, University of Minnesota, for many helpful suggestions in the conduct of this work. Appreciation is also due to Edward J. Solheim and Byron Olson who conducted the tests, performed calculations and prepared the figures. The assistance of Miss E. Thorson in typing the manuscript is also appreciated.

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Measuring Repeated Strains with a Modified Null-balancing Strain Indicator

Accurate measurements can be made within certain amplitude and frequency limits if proper filter is added to the measuring circuit

by R. S. Loubser and W. W. Schroeder

Strain indicators of the null-balancing type, e.g., the Baldwin Type M, are designed mainly for accurate measurement of static strain. Although the unrectified bridge output can also be displayed on an oscilloscope screen when the strain is dynamic, it is not possible to determine the exact balance point when dynamic strain is to be measured. Dynamic measurements are therefore inherently less accurate and convenient.

The reasons are: (1) Even if the bridge is balanced both resistively and capacitively, the second harmonic of the carrier frequency is still present in the bridge output. (2) There is no indication of the phase of the signal.

The advantage of the null method used for measuring static strains by observing the output of the phase-detector on galvanometer G (Fig. 1) could be retained for measurements of repeated dynamic strains if the following amendment is incorporated in the instrument, as applied to a Baldwin Type M strain indicator.

The phase-detector output is displayed on a d-c oscilloscope after filtering out the 1-kc carrier frequency. The oscilloscope trace now replaces the galvanometer needle as the "null" indicator. The filter as shown in Fig. 1, causes a 20-db/octave attenuation above 200 cps, the total noise level on the trace being equivalent to approximately ±1 microin./in. strain. An output of 6 mV/1 microin./in. unbalance of the indicator is obtainable if the circuit is capacitively balanced, i.e., ±2 microin./in. can be easily resolved. The influence of capacitive unbalance is mainly to reduce the sensitivity of the indication. A 200 pF unbalance represents a limit for practical use in this manner. The tendency for zero drift is approximately equal to that using the meter as a balance indicator. A small change in null-balance setting on the indicator occurs when changing from meter balance to the proposed balance method, but this change is constant for a given test condition. The balance point on the oscilloscope can be conveniently checked by switching off the indicator.

If a regularly repeated signal is applied to the indicator and balanced at a peak value, a trace of the type shown in Fig. 2 is obtained. A similar balance value can be determined for the maximum peak. It will be observed that the signal is badly distorted due to amplifier saturation during that part of the signal where the unbalance is excessive. For accurate presentation of peak balance values, the amplifier must recover before each peak is reached. The limiting saturation condition up to which the method is useful depends on the signal amplitude, frequency and wave shape, and also to some extent on the bridge resistance and capacitance unbalance; these should be determined for each indicator under its particular working conditions. The average recovery time for a single sudden change in strain is 3 millisec, so that a frequency of 100 cps represents an upper working limit in any case. For the Type M strain indicator used, practical limits are approximately 3000 microin./in. strain at 1500 cpm for a sinusoidal signal, the accuracy attainable being approximately $^1/_2$%. For larger amplitudes, a gage desensitizing circuit as recommended by the manufacturers* should be used.

R. S. Loubser and W. W. Schroeder are associated with South African Council for Scientific and Industrial Research, Pretoria.

** Baldwin Instruction Book Bull. 312-A, Baldwin-Lima-Hamilton Corp. Waltham, Mass.*

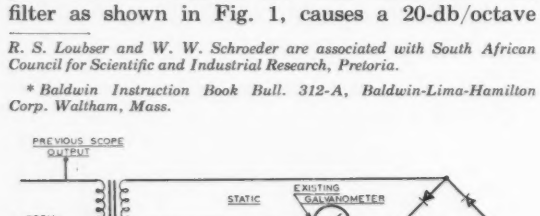
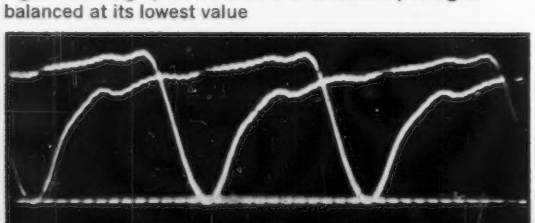


Fig. 1—Filter unit applied to Baldwin strain indicator

Fig. 2—Oscillographic trace of a sinusoidal input signal balanced at its lowest value



Conclusion

Repeated dynamic strains up to certain amplitude and frequency values can be measured with a sensitivity equal to that attainable statically, if an appropriate filter is added to the measuring circuit. The method appears particularly valuable if the strain-indicator amplifier has a long linear range such as the new Baldwin Model N strain indicator where both wave shape and amplitude could be studied.

Summary

The output of the phase detector of a Baldwin Type M strain indicator can be filtered and displayed on a d-c oscilloscope which acts as the null detector. Repeated strains can then be accurately measured within certain amplitude and frequency limits.

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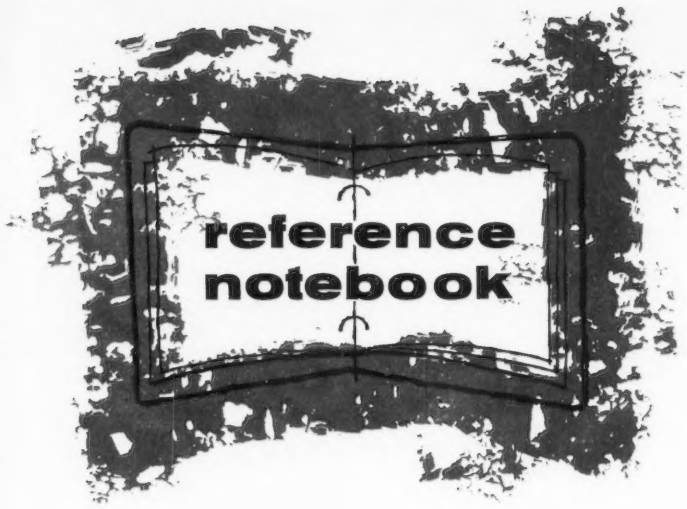

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sesa news

Tentative Program Announced for Philadelphia Spring Meeting

Following the release by the Papers and Proceedings Committee of the technical papers selected for presentation at the SESA 1961 Spring Meeting, Stewart W. Sennet, general chairman, announced tentative program details. A schedule of 16 presentations by authors of outstanding papers was selected as the principal event of the meeting being held this year on May 10–12 at the Benjamin Franklin Hotel, Philadelphia. Four additional papers were solicited by the local committee for the particular purpose of developing the meeting theme, "Experimental Mechanics at High Temperatures." The total of 20 papers has been arranged in five sessions to be held each morning and afternoon of the three-day period, except Thursday afternoon, May 11th, when two plant tours have been scheduled.

In addition to the highly interesting variety to be found in the excellent papers selected for presentation, a full program of other activities has been announced. Responsibility for these activities, as well as all the local arrangements, has been assumed by the Delaware Valley Section, this year's host for the Spring Meeting. Progress to date gives every indication of a highly successful affair.

Among the highlights of the busy program are to be found a Wednesday evening seminar on "The Role of Computers as a Tool in Experimental Mechanics," two interesting plant tours for Thursday afternoon, a social hour and banquet for Thursday evening and a full program of ladies' activities designed to offer ample diversion for the distaff side.

Technical Papers

The technical papers should arouse a wide interest. Their titles, authors and time of presentation are given in the full program to be found in the following pages. The 16 papers selected by the Papers and Proceedings Committee will appear in forthcoming issues of EXPERIMENTAL MECHANICS and will subsequently be published in the bound volumes of the SESA *Proceedings*. The presentations made in the Theme Session must be presented to the Papers Committee for acceptance before becoming eligible for publication by SESA.

Seminar on Computers

The purpose of the Wednesday evening seminar is to introduce engineers to the possibilities offered by analog and digital computers in the field of experimental mechanics. Many engineers tend to avoid the use of computers because they do not know what can be done. It is hoped that the seminar will arouse sufficient interest on the part of the listeners so that they will pursue the further training necessary for use of such equipment.

One speaker will cover each of the two types of machines, analog and digital. An introductory talk will explain the basic kind of information each of the machines is inherently capable of handling, how the data are introduced and in what form the answer to a problem will appear. This will be followed by an illustration of the working of an actual problem in experimental mechanics. Time will not permit a visit to actual installations nearby, as was previously hoped. Rather, the problem solutions will be demonstrated with the aid of slides.

Plant Tours

Two tours of unusual interest have been arranged for Thursday afternoon. One tour is to the well-known Eddystone Power Station of the Philadelphia Electric Co. The other will cover General Electric Co.'s Space Science Laboratory. The Eddystone Power Station is the first of its kind to operate in the high-temperature, high-pressure ranges of 5000 psi and 1200° F. Its many unusual features and the difficult problems encountered in its design will be of interest to all engineers familiar with efforts to attain higher efficiency in generating equipment.

At the Space Science Laboratory visitors will have the unique opportunity for viewing the latest equipment being used in the pioneering science for the conquest of space.

Social Hour Precedes Banquet

On Thursday evening a "dutch-treat" social hour will afford members and guests a welcome opportunity to renew old acquaintances and to make new ones in the pleasant atmosphere of the Betsy Ross Room. Following the social hour, the Banquet will be held in the Crystal Ballroom. An eminent speaker has been invited to address the audience on a subject of timely interest. For the present, the committee is keeping his identity under wraps in order to obtain more specific confirmation before releasing the details. An excellent repast has been arranged by the Banquet Committee.

Attractive Ladies' program

Special efforts have been made this year to develop a Ladies' Program that will provide the right amount of diversion and entertainment for the wives who will accompany their husbands to the Spring Meeting. The three-day program includes two days of sightseeing in and around historic Philadelphia and one day for shopping or rest.

The first trip on Wednesday will include visits to Independence Hall, Betsy Ross House and other points of early American fame.

On Thursday, a late "brunch" will signal the start of a trip to Longwood Gardens, the famous estate built by Pierre S. du Pont. The extensive grounds of the estate contain one of the most complete and unusual horti-

cultural displays in the country and houses what is believed to the largest orchid collection extant.

No official program has been set for Friday. This will give the ladies opportunity to rest, shop or arrange their activities at pleasure.

Other Events

Many other events will compete for the attention of those who attend the meeting, especially for those members who will be engaged in SESA business. Each morning will see the authors up bright and early for the Authors' Breakfast where last minute remarks will help launch the presentations to follow. An exhibit open all day for the benefit of visitors will include various educational displays. Meetings of the standing committees of SESA, the Committee's Dinner and the Executive Committee Meeting will be held in the Poor Richard Room on Wednesday afternoon and evening. Registration will be handled each day from 8:30 A.M. to 4:00 P.M. except Friday when noontime will be the limit. For the convenience of early arrivals, the booth will also be open Tuesday from 7:30 P.M. to 10:00 P.M.

Except for the Authors' Breakfasts and the Committee Meetings, all activities of the 1961 Spring Meeting have been arranged in and around the Crystal Ballroom so as to provide maximum convenience and economy of movement.

Employment Service Bulletin . . . a Service Feature for Members

As a service to SESA membership—both corporate and individual—a column will regularly carry information regarding "Situations Wanted" and "Positions Available." Members are encouraged to send a brief of particulars to Experimental Mechanics, 21 Bridge Square, Westport, Conn. An insertion will appear once and will be limited to 75 words. No charge will be made for this service.

Local Committee Members Named for First International Congress

The many local activities concerned with the successful launching of the SESA 1961 Annual Meeting—this year titled, The First International Congress on Experimental Mechanics—got off to an official start recently with the appointment of the Arrangements Committee. The International Congress will be held in New York City, November 1–3, at the Hotel New Yorker and will follow by a day the International Symposium on Photoelasticity to be held in Chicago.

As announced by E. R. Lewis, chairman of the New York Section, the members who will head up the various activities are as follows: General Chairman, V. L. Salerno; Vice Chairman, D. L. Pasquine; Educational and Technical Program, S. V. Nardo; Exhibition, T. Braunstein; Secretary-treasurer, S. Black; Technical Papers, E. Miller; Ladies' Program, Mrs. D. Pasquine and Mrs. S. Black. Other appointments will be announced as they are made.

Considerable ground work is presently under way at the local, national and international levels. At the local level, the selection of the entire mezzanine floor of the Hotel New Yorker as the site for the 1961 Congress signaled the start of activities. A factor in this consideration was the convenient layout of rooms which will permit easy access all on one floor to and from points of registration, technical sessions and exhibits. It is believed that this arrangement will not only benefit those attending the Congress but will encourage greater attendance at the exhibit. Because of the international nature of the meeting and the location in New York City, it is expected that the Exposition will be the largest and best attended in SESA history. Many manufacturers, research groups and educational institutions are expected to take advantage of the unusual opportunities offered by the International Congress on Experimental Mechanics.

At the discussion stage is a suitable theme for the meeting and the possible solicitation of papers for a Theme Session. Also under consideration are the possibilities of plant tours, ladies' programs, banquet, banquet speaker, symposium, the specific arrangement of the program and other details.

On the national level, Roscoe Guernsey, chairman of the SESA Papers and Proceedings Committee, reports that already a greater number of papers has been received than is normally presented at an annual meeting. This fact is a good barometer of the intense interest that is building up for the Congress.

Meanwhile, activity on the international level has borne fruit. The Organizing Committee for the International Congress on Experimental Mechanics, under D. C. Drucker and W. M. Murray as co-chairmen, has sent out a number of invitations for participants from other countries. Many of these have responded favorably.

International Symposium on Photoelasticity Aided by Army Grant

The Illinois Institute of Technology has received a grant of $9950 from the Office of Ordnance Research, U. S. Army, for the purpose of organizing and conducting the International Symposium on Photoelasticity.

Formulation of the proposed program and contract negotiations were made by M. M. Frocht, IIT professor of mechanics and director of experimental stress analysis, whose research has been sponsored by the OOR for a number of years. The symposium will be held at Illinois Tech on Oct. 29–31, 1961. The only other meeting of this kind was held in Brussels, Belgium, in 1954.

Scientists from the U. S., England, France, Sweden, Switzerland, Germany and Japan are expected to present papers.

According to Dr. Frocht, the symposium's chairman, "the program will be comprehensive and will embrace basic aspects, such as two- and three-dimensional photoelasticity, photoplasticity, dynamic photoelasticity and photothermoelasticity.

"The aim will be to encompass on an international scale the fundamental research activities in these areas, as well as significant technological applications," he said.

Besides IIT and the Army Ordnance Research Office, other cosponsoring organizations are the SESA, Office of Naval Research, the Office of Scientific Research, U. S. Air Force, the Applied Mechanics Division of the American Society of Mechanical Engineers and the ASME Chicago chapter.

Be an Active Member . . .

. . . Support your Society!

TENTATIVE PROGRAM

1961 SESA SPRING MEETING

MAY 10-12

BENJAMIN FRANKLIN HOTEL

PHILADELPHIA, PA

MAY 10—WEDNESDAY MORNING

8:00 A.M. Authors' Breakfast
Robert Morris Room

8:30 A.M. Registration
Crystal Ballroom Foyer

8:30 A.M. Exhibit Opens
Crystal Ballroom Foyer

9:30 A.M. Words of Welcome
Crystal Ballroom

Technical Papers Session
9:45 A.M. *Crystal Ballroom*

Residual Stresses in Shot-Peened Aluminum Bars
by J. T. Norton, Massachusetts Institute of Technology, and M. J. Donachie, Jr., Electric Boat Div., General Dynamics Corp.

Measure of Residual Stress in Disks from Forgings
by D. Mack, General Electric Co.

Fatigue Testing of Thin Tapes
by H. Pih, Marquette University and A. T. Shalkey, International Business Machines Co.

A Strain-gage Technique for Measuring Internal Strains of Loaded Artificial Solids (By Title Only)
by S. Serdengecti, California Research Corp., J. A. Hendrickson and L. Skjelbreia, National Engineering Science Co.

MAY 10—WEDNESDAY AFTERNOON

4:00 P.M. Meetings of Standing Committees
Poor Richard Room

6:30 P.M. Committees' Dinner
Poor Richard Room

8:00 P.M. Executive Committee Meeting
Poor Richard Room

8:00 P.M. Seminar
Valley Forge Room

Technical Papers Session
2:00 P.M. *Crystal Ballroom*

Elastic Stability of Circular Cylindrical Shells Stabilized by a Soft Elastic Core
by William S. Goree and William A. Nash, University of Florida

Effect of a Transverse Temperature Gradient on Column Strength
by J. E. Winkler, Space Technology Laboratories

A Two-millisecond Rise Time One-tenth Percent Noise Level Rotational Speed Measuring System
by P. Stein, Editor, STRAIN GAGE READINGS

Computing the Velocities and Displacements Associated with Transient Accelerations by Integrating the Response of Accelerometers
by Robert W. Kearns, Kearns & Law

MAY 11—THURSDAY MORNING

8:00 A.M. Authors' Breakfast
Robert Morris Room

8:00 A.M. Registration
Crystal Ballroom Foyer

8:30 A.M. Exhibit Opens
Crystal Ballroom Foyer

Theme Session
"Experimental Mechanics at High Temperatures"
9:00 A.M. *Crystal Ballroom*

Thermal Fatigue
by T. C. Yen, Franklin Institute Laboratories for Research and Development

High-temperature Tension and Creep Tester for Wire
by B. R. Sinwell, Westinghouse Electric Corp.

Experimental Study of Dynamic Plasticity at Elevated Temperatures
by James F. Bell, The Johns Hopkins University

Accuracy of Thermocouples in Radiant-heat Testing
by Francis J. White, Naval Air Material Center

MAY 11—THURSDAY AFTERNOON

1:00 P.M. Busses Load for Plant Tours to the Eddystone Power Station of the Philadelphia Electric Co. and the Space Science Laboratory of General Electric Co.

6:00 P.M. Social Hour
Betsy Ross Room

7:00 P.M. Banquet
Crystal Ballroom

MAY 12—FRIDAY MORNING

8:00 A.M. Authors' Breakfast
Robert Morris Room

8:30 A.M. Registration
Crystal Ballroom Foyer

8:30 A.M. Exhibit Opens
Crystal Ballroom Foyer

Technical Papers Session
9:00 *A.M. Crystal Ballroom*

Fringe Multiplication and Thickness Effect in Birefringent Coatings
by E. E. Day, A. S. Kobayashi and C. N. Larson, University of Washington

A Photographic Method to Sharpen and Double Isochromatic Fringes
by J. W. Dally and F. J. Ahimaz, Cornell University

A Photoelastic and Strain-gage Investigation of the Penetration of Thin Plates
by W. Goldsmith and L. Dabaghian, University of California

Experimental Means of Analyzing Stresses and Strains in Rocket Propellant Grains
by A. J. Durelli, Armour Research Foundation

MAY 12—FRIDAY AFTERNOON

Technical Papers Session
2:00 P.M. *Crystal Ballroom*

Extending the Stress Analysis Range of Brittle Coatings
by V. J. Parks, K. E. Hofer and A. J. Durelli, Armour Research Foundation

Calibration of Stresscoat by Impact
by G. W. Brown and D. M. Cunningham, University of California

Stress Analysis of HNPF Loading-face Shield
by R. D. Chipman, Allegany Instrument Co., M. D. Lynch and R. M. Peterson, Atomics International

A Test Method for the Mechanical Evaluation of Gas-turbine Blades in an Actual Centrifugal Field
by R. F. French, International Harvester Co.

BOUND VOLUMES OF PROCEEDINGS AVAILABLE

Still available are a number of bound volumes of the series **Proceedings of the Society for Experimental Stress Analysis.** Published twice yearly, these volumes comprise the technical papers presented at national meetings and contain the results and descriptions of analytical and experimental investigations, as well as discussions of technique and instrumentation studies.

Price per copy: $7.00 to nonmembers and $6.00 to members; plus postage. Copies may be ordered through the Society for Experimental Stress Analysis, 21 Bridge Square, Westport, Connecticut.

SPEAKERS RESPONDING TO CALL

In response to the recent invitation extended to all qualified speakers who are willing to address local SESA section meetings without charge, a number of volunteers have so far offered their services. Additional speakers are being sought and those who believe that they have information of current interest to members are urged to send particulars to Clarence R. Smith, Chairman, SESA Local Sections Committee, Mail Zone 63-10, Convair, P. O. Box 1950, San Diego 12, Calif., or to SESA Headquarters, 21 Bridge Square, Westport Conn.

Among the topics selected by the speakers to date are: "Static Testing—Opening the Door into Space"; "The Role of Materials in Experimental Mechanics"; "High Temperature Structural Testing of the X-15"; "Acoustic Emission Under Applied Stress"; "Low Cycle Fatigue"; "The Cause and Prevention of Fatigue Failures in Metallic Structures."

Advance Registration Form

SOCIETY FOR EXPERIMENTAL STRESS ANALYSIS
1961 Spring Meeting—May 10, 11, 12, 1961
Benjamin Franklin Hotel, Philadelphia, Pa.

Name and home address:

Company affiliation and address:

_____

_____

_____

Registration Fees

At meeting:	Advance Registration:	
$6.00 member	$5.00 member	$_____
8.00 nonmember	7.00 nonmember	_____
1.00 student	1.00 student	_____

Banquet—Thursday evening, May 11
(Speaker and Topic—To be announced) $6.50 per person _____

Plant tours—Thursday P.M., May 11 **Bus fare per person:**

Eddystone Power Plant $1.50 _____
G. E. Space Sciences Laboratory 1.50 _____

Will your wife accompany you to Philadelphia? Yes___ No___

Total enclosed $=====

Check or money order payable to: "1961 Spring Meeting SESA". Please return form with payment to: **Henry Cook, Instruments Division, Budd Co., P. O. Box 245, Phoenixville, Pa.**

section reports

Delaware Valley

Turbine-blade Vibrations

The winter meeting of the *Delaware Valley* Section was held on February 1st. Following dinner, a business meeting and technical session were held in the Development Laboratory of the Westinghouse Electric Corp.'s Steam Turbine Plant.

Stewart W. Sennet, general chairman of the SESA 1961 Spring Meeting which is to be held in Philadelphia on May 10th, 11th and 12th, reported that plans for the meeting were progressing well. He expressed appreciation for the excellent cooperation and work of the local section members to date and asked continued efforts by all to make this one of the best meetings on record. He particularly urged members to persuade their wives to participate and make the women's program a success.

A technical session entitled "Measurement of Stresses and Vibrations in Turbine Blades During Operation" was presented by several Westinghouse engineers. D. R. Rupp, Senior Engineer in Medium Turbine Engineering, served as moderator and spoke of the historical development of the instrumentation system now in use. A. J. Partington of Development Engineering described certain problems in designing turbine blades which lead to experimental testing and showed how the measurements taken help solve them. Techniques of strain gage installation and protection against the effects of wet steam, and significant details of slip rings and strain-gage circuits were covered by D. W. Darkow, also of Development Engineering. An elaborate set of instruments, described by V. Donato of Large Turbine Engineering, included multichannel magnetic tape recording of strain-gage signals, multiplexing of signals, analog plotting of output data and automatic digital printing of data. In the last talk, F. Michel, Industrial Gas Turbine Engineering, told of a test of a gas turbine in which turbine blade vibrations were measured with strain gages operating at temperature near 1000° F. Following the talks, the instrumentation was demonstrated and a typical test was "rerun" via multichannel magnetic tape recording. The facilities of the Westinghouse Steam Division Development Laboratory were open for inspection.

One hundred and twenty-one interested persons braved near zero temperatures to attend the meeting.

(Reported by R. Moser for Delaware Valley Section)

New England

History of Stress Analysis

On Feb. 1, 1961, the *New England* Section held a joint meeting with the Society for Nondestructive Testing at the MIT Faculty Club. The speaker was Frank Tatnall, the highly respected "grand gentleman" of stress analysis. Mr. Tatnall talked on the history of stress analysis and strain gages. His extremely interesting speech went from the carbon disk

SESA Section Meeting Calendar

April 3

SALT LAKE CITY Section. Organizational meeting. Talk: "The Cause and Prevention of Fatigue Failures in Metal Parts," by C. R. Smith, design specialist, General Dynamics Corp., San Diego, Calif. Contact: G. W. West, Engineering Laboratory, Thiokol Chemical Corp., Brigham City, Utah.

April 4

DENVER Section. Organizational meeting. Talk, "The Cause and Prevention of Fatigue Failures in Metal Parts," by C. R. Smith, design specialist, General Dynamics Corp., San Diego, Calif. Contact: M. Barrett, Ketchum, Konkel & Hastings, Engineering Consultants, Denver, Colo.

April 11

SEATTLE Section. Organizational meeting. Talk, "The Cause and Prevention of Fatigue Failures in Metal Parts," by C. R. Smith, design specialist, General Dynamics Corp., San Diego, Calif. Contact: Prof. Emmet Day, Department Mechanical Engineering, University of Washington, Seattle, Wash.

April 20

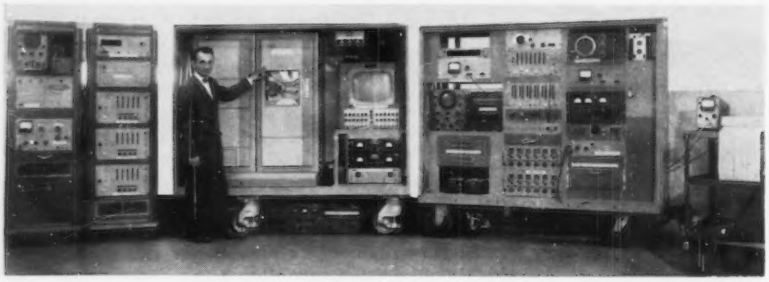
SOUTHWEST Section. Roger Young Auditorium. Talk: "The Static Test of the Simon Rodia's Towers," by B. Goldstone; also, "Applications of Photostress in the Atlas Program," by W. Witzell.

April 27

HUDSON-MOHAWK Section. Rensselaer Polytechnic Institute, Troy, N. Y. Dinner: 7:00 P.M. Talk.: 8:00 P.M. "Piezoresistive Strain Gages," by Dr. W. P. Mason, Bell Telephone Laboratories, Murray Hill, N. J.

EDITOR'S NOTE: Notices to be published in any one issue must reach SESA headquarters prior to the 15th of the second previous month. Please give full information concerning date, time, place, topic and speaker for each meeting.

INSTRUMENTS FOR TURBINE-BLADE STUDIES



Instruments used to measure turbine blade vibrations and stresses are described by Vince Donato at the technical session of the Delaware Valley Section held on February 1st at the Westinghouse Steam Turbine Plant

strain gages through wire gages, photo stress and ended with the very latest semiconductor gage.

The presentation was livened considerably by his interjections of philosophy. He also gave an intimate personal glimpse of his association with such people as Prof. DeForrest, Mr. Simmons and Professor Ruge all of whom had a hand in developing the wire strain gage. He also mentioned many other famous men in the field with whom he had close association. Everyone in the room was held in rapt fascination, and many could have gone on listening to him indefinitely.

Many of the leading stress-analysis people and friends of Mr. Tatnall in the local area attended and helped the presentation by adding to glimpses of the history.

All in all, it was one of the most fascinating technical meetings which the Section has held.

(Reported by B. P. Friesecke for the New England Section)

Southern Ohio

Data-handling System

On Wednesday, Jan. 18, 1961, the Section held its third meeting of the 1960–61 season at Wright-Patterson Air Force Base, Ohio.

A total of 32 members and guests attended the meeting held at the Air Force Elevated Temperature Standard Test Facility, which is located on the base.

The meeting began with a short talk by R. L. Cavanagh on the operation of the facility and its mission in the space age. Special emphasis was placed on the newly installed high-speed data-handling system and its digital computer, which is believed to be the largest of its kind, having a total capacity of more than 1900 channels for measurements of strain, temperature, load and deflection.

Upon completion of the talk, all present were given a guided tour of the test facility and a demonstration of the computer and integrated system. Everyone was very much impressed with the speed of the system and the enormous volume of information it is capable of recording and storing. This tour was made possible through the efforts of F. E. Hussong and R. L. Cavanagh, who were ably assisted by other members of the staff of this facility.

(Reported by J. H. Jaspers for the Southern Ohio Section)

Southwest

Structural Testing the X-15

The February 10 meeting was called to order by the Chairman, G. D. Lindeneau. Chapter business included the announcement of membership cards to be issued for the first time to the local section for 1961–62 period. Mr. Vanbebber and Mr. Ormand announced they would donate a public-address system to the chapter at the April meeting.

The chair was then turned over to the speaker, O. L. Gillette, supervisor of instrumentation and elevated-temperature testing techniques, North American Aviation, Inc., Los Angeles Division. Mr. Gillette presented a paper on "High-temperature Testing of the X-15." A brief of his talk follows:

"In 1955 the Structures Lab-

TATNALL ADDRESSES NEW ENGLAND SECTION

Jovial Frank Tatnall rises to address the February 1st meeting of the New England Section. His fellow "old timer," National Treasurer Bill Murray, is shown in background

This is a partial view of the many members and guests who turned out to hear Mr. Tatnall speak on "The History of Stress Analysis." Executive Committee member Ferdi Stern and former Proceedings Editor "Chuck" Mahlmann are among those enjoying Frank's "quips"

TESTING OF X-15 DISCUSSED AT SAN DIEGO MEETING

Part of the 71 members and guests of the Southwest Section who attended the February 10th meeting to hear O. L. Gillette talk on the "High-temperature Testing of the X-15 Aircraft." The scene is laid at the Admiral Kidd Club, San Diego, Calif.

At the speakers' table the genial faces are those of, left to right: C. R. Smith, program chairman; F. N. Singdale, vice-chairman; O. L.Gillette, speaker; G. L. Lindeneau, chairman; E. K. Winslow, secretary-treasurer. Lower right, B. J. Wolfe, membership chairman

oratory began preparation for the structural testing of the X-15. This program was to last through 1958. Two methods for testing structural components were available at this time, (1) heat flux and (2) temperature to flight profile, precalculated. The temperature to flight profile method was selected. The following testing techniques were developed: (1) paints to produce desired heat absorption, (2) liquid nitrogen and air flow to produce the desired thermal gradients on the inside of the structure, (3) grid lines painted on structure to determine bulking, (4) ceramic rods to measure deflection, (5) gold plated reflectors to obtain 2200° F for 250 sec with quartz lamp banks, and (6) compression loads by cables run through specimen to loaded surface. Strain-gage data at temperature was limited in 1956 due to availability of reliable strain gages. Iron-con-

SYMPOSIUM—WORKSHOP
on
VIBRATION, ITS MEASUREMENT AND CONTROL
at
National Bureau of Standards
East Lecture Room
on
THURSDAY, MAY 4, 1961—9:30 A.M.–4:00 P.M.

sponsored by
WASHINGTON AREA SECTIONS
INSTRUMENT SOCIETY OF AMERICA
SOCIETY FOR EXPERIMENTAL STRESS ANALYSIS

Planned program includes:

- **VIBRATION MEASURING INSTRUMENTS**
- **SIMULATION OF FLUTTER**
- **ANALOG SIMULATION OF VIBRATION**
- **DESIGNING AGAINST VIBRATION**
- **CALIBRATION OF VIBRATION MEASURING SYSTEMS**
- **MEASURING THE VIBRATION OF BUILDINGS**
- **SEISMOLOGY**

Participants include

Dr. I. Vigness, Dr. H. M. Trent, Fr. B. McConnell, D. R. Tate, B. Reznek, J. P. Kearns, H. Forkois

Interested persons are cordially invited to attend. To facilitate planning and to expedite registration, advance notice of attendance giving name, organization and address should be sent to:

R. L. Bloss
3028 Industrial Building
National Bureau of Standards
Washington 25, D. C.

NO REGISTRATION FEE

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For details, circle No. 9 on Reader Information Card

NEW MICRODOT STRAIN GAGE WELDS TO ALUMINUM

To meet the demand for a weldable strain gage designed for attachment to aluminum missile skins, Microdot Inc. has developed a new edition to their basic weldable strain gage line.

The new gage, which is designated the AL-E series, can be attached to almost all aluminum alloys. Each gage will operate at temperatures equal to the deterioration point of aluminum itself (750°F for short periods is acceptable). Normal gage factor for the "aluminum gage" is 1.75.

These gages are $^3/_{16}$" wide and either $1^5/_{32}$" or $^9/_{16}$" long. Both 60 and 120 ohm gages are offered. All types can be inherently temperature compensated for aluminum alloys.

The "aluminum gage" can be attached to aluminum with an 80 watt-second capacitor discharge weldor, using RWMA electrodes in the hand piece. Special cleaning and welding instructions are furnished by Microdot.

In addition to the "aluminum gage", Microdot manufacturers a line of standard weldable strain gages, hermetically sealed load cells with adjustable temperature sensitivity, and temperature sensors. These strain gage devices are designed and manufactured by the Instrumentation Division of Microdot Inc. which also offers transducers, amplifiers, power supplies, strain gage signal conditioning equipment, rf cavities, oscillators, and telemetry transmitters.

MICRODOT INC.

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South Pasadena, California
MUrray 2-3351 SYcamore 9-9171

For details, circle No. 10 on Reader Information Card

stantan thermocouples welded to structure were found to be very dependable for temperature measurement. Temperatures were recorded by commutation, 20 per channel, 120 recordings. In actual testing, thermal gradients through the structural cross section were found to be very difficult to obtain whereas skin temperatures were easily obtained. This was felt to be typical of structure due to variances in joint tolerance. Area control was best for small areas, therefore, components were tested. The T-3 heat lamp was excellent for creating desired temperatures. The ends of the lamps were cooled. Heat program control was obtained by ignitron tubes. These tubes gave the best control and were economical. Strain measurements gave the greatest problem. At the time of the testing no solution for accurate strain measurement was found.

"The success of the test program was substantiated by the successful flight of the X-15 aircraft. A color movie of this flight was shown. Also, color movies of the actual heat tests and equipment was shown."

The meeting was then adjourned.

The Southwest Chapter is unanimous in its thanks to Mr. Gillette and North American Aviation for the presentation of this fine paper.

The next meeting of the Southwest Chapter will be held in Los Angeles, Apr. 20, 1961, at the Roger Young Auditorium. At this time Budd Goldstone will present a paper on "The Static Test of the Simon Rodia's Towers" and Bill Witzell will present a paper on "Applications of Photostress in the Atlas Program."

(Report and photographs by E. K. Winslow for Southwest Section.)

Washington Area

Impedance, A Challenge and Do-it-yourself

Twenty-one members and guests ignored impending snow warnings on February 1st to attend the Section's third general meeting of the season. The scheduled speaker, on the other hand, was unable to appear because of illness, but the audience was amply rewarded for their loyalty by pinch-hitter Fred Schloss of the David Taylor Model Basin, who spoke on "Measurement of Mechanical Impedance." Defining mechanical impedance as the ratio of force amplitude to velocity amplitude, Mr. Schloss reviewed the development of a piezoelectric device for measuring impedances of submarine hulls in the frequency range of 20 to 5000 cps, at acceleration levels down to only 10^{-5} g.

The speaker went on to describe instrumentation for evaluating the phase difference between the force and velocity vectors. The principle involves a feedback, at 0 or 90 deg, from the acceleration signal to the force signal. The magnitude of the feedback is adjusted until the resultant force vector is minimized, at which time the two vectors are exactly 90 deg out of phase.

Following a coffee break, the meeting reconvened for the showing of two interesting films, "Challenge at Carquinez" and "Lifetime Protection." The first film described the design and construction of the first all-welded steel bridge at Carquinez Straits in California. The second film had nothing to do with insurance, but rather depicted the installation and advantages of polystyrene foam as a moisture-resistant structural insulation for residential basements.

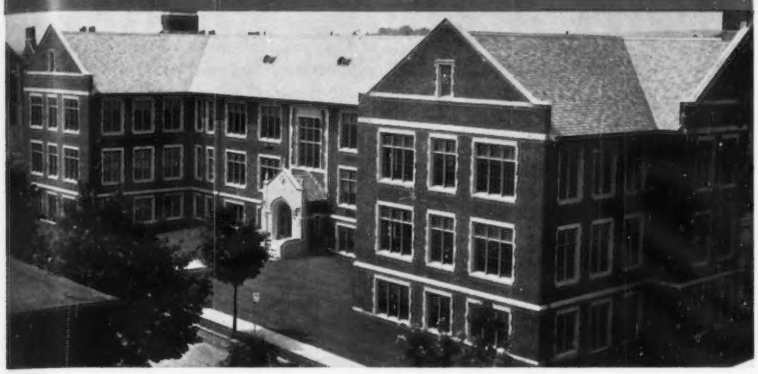
(Reported by L. Mordfin for Washington Section)

Representatives in Organizing Areas

India (West Bengal)	Prof. C. N. Lakshminarayana Mechanical Engineering Department Indian Institute of Technology Kharagpur, West Benga
Japan (Tokyo)	Dr. Mineo Yamamoto Tokyo Agricultural and Technical University Mechanical Department, 151, 2-Chome, Nakamachi Koganei City, Tokyo, Japan
Switzerland (Lucerne)	Max Buri Eidg. Flugzeugwerk Emmen Emmen (Lucerne) Switzerland

Plan Now To Attend
1961 Spring Meeting
May 10 - 12
Benjamin Franklin Hotel
Philadelphia, Pa.

from the campus



Higgins Laboratory, Worcester Polytechnic Institute

SESA MEMBERS TO LECTURE AT MIT

Members of the Society will pool their knowledge in cooperation with MIT in presenting three short courses for engineers, physicists and metallurgists who are currently engaged in industry.

SESA Past President Milton M. Leven, of the Westinghouse Research Laboratories, and Felix Zandman, director of research at the Instruments Division of the Budd Co., will collaborate with Jerome Catz and William M. Murray of the Department of Mechanical Engineering at MIT in presenting the course, Industrial Photoelasticity. This course, given June 19–23, is intended for beginners, and will include a review of classical two-dimensional photoelasticity together with detailed discussions of the latest techniques involved in three-dimensional applications and the uses of photoelastic surface coatings. Laboratory exercises to supplement the lectures will be included.

During July, William T. Bean, Jr., consulting engineer of Detroit, Given A. Brewer, chief engineer of Brewer Engineering Laboratories, in Marion, Mass., Greer Ellis, of Ellis Associates, in Pelham, N. Y., Peter K. Stein, editor of *Strain Gage Readings*, Phoenix, Ariz., and Leon J. Weymouth of the Electronics & Instrumentation Division of the Baldwin-Lima-Hamilton Corp., of Waltham, Mass., will join Dr. Catz and Dr. Murray, and other members of the MIT staff in presenting two 1-week programs on strain gages.

The first of the two strain-gage courses will be given July 10–14 and will consist entirely of lectures. The topics to be discussed will include the basic operating principles and characteristics of wire, foil and semiconductor strain gages; installation techniques for high, low and medium temperature levels; electric circuits and instrumentation; static and dynamic strain measurements; interpretation of strains into terms of stress via rosette analysis; measuring systems; measuring instruments; and the use of multiple gage bridges for the measurement of force, moments, torque, thrust, shear, pressure, etc.

The second course on strain gages will be given July 17–21 and will be devoted entirely to the laboratory. This will include a number of experiments which have been specially designed to illustrate the theory presented during the previous week. Special attention will be paid to the techniques of gage installation, both for normal and high temperatures, electrical instrumentation and various applied mechanics aspects of strain gage applications.

In addition to the formal parts of these programs, certain recreational and social activities are being organized. The chief function of this nature will be a luncheon and informal exhibit of equipment sponsored by the Society; this will be held about the middle of the first week on strain gages.

Further information on all aspects of the MIT photoelasticity and strain-gage courses may be obtained by writing to Dr. W. M. Murray, Room 3-260, Massachusetts Institute of Technology, Cambridge 39, Mass.

STRAIN GAGE COURSE IN TEXAS

SESA will collaborate with Southwest Research Institute in joint sponsorship of a short course on strain gages, at San Antonio, Tex., April 17–21, 1961.

Members of the society are planning a week's program which will consist of a series of illustrated lectures and optional, supplementary laboratory. The course is intended for stress analysts, designers, laboratory workers and supervisors, with a bachelor's (or higher) degree in mathematics, physics, engineering, or metallurgy. However, in certain cases, experienced technicians without degrees can be expected to gain full benefit from the program.

A broad picture of the theoretical and practical considerations involved in the industrial uses of strain gages will be presented. Special attention will be directed toward applications at high and low temperatures and to new developments including semiconductor gages.

Topics to be discussed will include the basic theory of resistance strain gages, characteristics of wire, foil and semiconductor gages, installation techniques for high, medium and low temperatures, electric circuits, instruments and instrumentation, measuring systems and the interpretation of strain observations into terms of stress.

The lectures will be presented by W. T. Bean, Jr., W. M. Murray, P. K. Stein, C. O. Vogt and L. J. Weymouth, all of whom are already well known not only as lecturers but also for their various publications related to strain gages.

M. M. Lemcoe, Chairman of the SESA South Central Local Section will represent both the Society and the Southwest Research Institute; in addition, he will direct all local arrangements. Detailed information may be obtained by writing to Dr. Lemcoe in care of the Department of Structural Research at the Southwest Research Institute 8500 Culebra Rd., San Antonio, 6, Tex., or by telephoning him (at San Antonio, GEneral 2-5221).

Employment Service Bulletin

As a service to SESA membership—both corporate and individual—a column will regularly carry information regarding "Situations Wanted" and "Positions Available." Members are encouraged to send a brief of particulars to Experimental Mechanics, 21 Bridge Square, Westport, Conn. An insertion will appear once and will be limited to 75 words. No charge will be made for this service.

X-RAY DIFFRACTION SUMMER COURSE

Illinois Institute of Technology is offering its fifth annual summer school in "X-ray Diffraction Analysis" this year, June 12–23.

Two one-week sessions provide training in the utilization of X-ray diffraction methods for the identification and characterization of crystallographic properties of commercial materials.

The "Powder Method of X-Ray Analysis" program, June 12–25, is designed to meet the needs of beginning students. "Advanced Methods of X-Ray Analysis," June 19–23, is offered to students who are already familiar with fundamentals of X-ray diffraction.

Each course is comprised of five days of lectures and laboratory, with special emphasis given to practical applications. No prerequisites are needed for the noncredit courses. Enrollment is limited to 16 students in each course.

Information concerning tuition fees and on-campus housing may be obtained from Prof. Leonid V. Azaroff, Illinois Institute of Technology, Technology Center, Chicago 16, Ill.

ENGINEERING SUMMER CONFERENCES

Twenty-seven one- and two-week courses are being offered this summer by the University of Michigan for scientists, engineers, and managers. Titled "Engineering Summer Conferences," the intensive courses are mostly from 5 to 10 days' duration and are scheduled during the months of June, July and August. Included in a wide range of practical, intermediate technical and advanced theoretical studies is a course on "Applications of Stress Analysis to Design and Metallurgy." Inquiries should be addressed to Engineering Summer Conferences, 126B West Engineering Building, University of Michigan, Ann Arbor, Mich.

SHORT COURSES AT GRADUATE LEVEL

Case Institute of Technology, Cleveland, Ohio, is offering a series of short courses intended to help professional engineers and scientists keep abreast of their fields. The special programs cover Operations Research, Digital Control Systems Engineering, Automatic Optimizing and Computing Control, Process Control Theory, Generalized Electrical Machine Theory and Analysis of Nonlinear Systems. Most of the courses are scheduled during June-July 1961. Inquiries should be addressed to: Herbert B. Schultz, Jr., Manager of Special Programs, Case Institute of Technology, University Circle, Cleveland 6, Ohio.

COLLEGIATE SCIENCE RESEARCH CONFERENCE

The University of Texas was host to about 120 students and faculty members from 23 Texas colleges at the Collegiate Science Research Conference, February 9–10.

The conference, sponsored by the National Science Foundation and the Texas Academy of Science, is one of four to be held at scientific research centers throughout the state.

Joseph J. Lagowski, University assistant professor of chemistry, conference chairman, said the purpose is to stress the nature, significance, requirements and rewards of present-day research in the natural sciences.

The students toured the University research laboratories in chemistry, physics, geology, botany, bacteriology and zoology, both on the main campus and at Balcones Research Center northwest of Austin. They conferred with University faculty members on careers in science and opportunities for graduate study and professional employment.

Conference speakers included Norman Hackerman, University Chemistry Department chairman and dean of research and sponsored programs; Stephen Clabaugh, geology professor, John R. Silber, associate professor of philosophy.

ASTRODYNAMICS COURSE OFFERED AT IIT

A new graduate course in astrodynamics will be offered by Illinois Institute of Technology for the first time during the 1961 spring semester.

The course program, designed to meet the needs of students who will be engaged in space technology, is concerned with problems of celestial mechanics and with such satellite problems in which the motion is far enough away from Earth so that gravity forces cannot be taken as a constant, but close enough so that air resistance affects motion.

Astrodynamics will be taught by Peter Chiarulli, professor of mechanics and department chairman.

Three Short Intensive Courses on

STRAIN GAGES & PHOTOELASTICITY

at the DEPARTMENT OF ENGINEERING
UNIVERSITY OF CALIFORNIA AT LOS ANGELES

FOR WHOM INTENDED

These three courses have been designed to present an integrated program on theory and practice, including a review of fundamentals and discussions of some of the latest techniques, especially those suited to high and low temperature strain gage applications. Attention will also be given to instrumentation, which, in many cases, can be used equally well with other kinds of transducers.

The courses are intended primarily for those with a bachelor's (or higher) degree in engineering, physics, or metallurgy. However, in certain cases, experienced technicians without degrees can be expected to gain full benefit.

The material content of the courses, although presented largely from the point of view of the mechanical engineer, has been selected to provide maximum benefit for those who are already skilled in stress analysis but who may have little experience in electrical instrumentation, and likewise for electronics experts who may not be quite so familiar with certain aspects of stress analysis.

STRAIN GAGE LECTURE COURSE

August 21 - 25, 1961 (Monday through Friday)

The first of the two strain gage courses will consist of a series of lectures supplemented with illustrative demonstrations. Some of these lectures have been prepared especially for beginners and consequently are of an elementary nature; however, other lectures of a much more advanced character will also be included in the series. The latter group will contain detailed information, especially relating to high temperature strain gages, which will be useful to those who already possess considerable experience. The content of the course has been selected and organized for maximum benefit to designers, metallurgists, laboratory workers, laboratory supervisors, theoretical stress analysts, and others, who wish to advance their knowledge relating to the uses of strain gages and the techniques which are available for producing optimum results.

Both the theoretical and practical aspects of strain gages themselves, methods of installation, and allied equipment, will be presented in the lectures. Details of strain gage applications for measuring force, torque, bending, shear, and accelerations, will be discussed as well as simple electric circuits for automatic computation of quantities which depend upon the simultaneous observation of strains at more than one location.

The following will be included in the topics which are to be presented:

Basic theory of Wire, Foil, and Semi-Conductor strain gages
Details of gage characteristics
Techniques for installing gages
Mechanical and Electrical aspects of the gage system
Fundamental Laws of Measurement
Electric Circuits for strain gages
Dynamic and Static Strain Measurements
Computing Bridge circuits for direct measurement of quantities depending upon several strains
Stresscoat as an Auxiliary Tool
Measuring systems and instruments
Low-temperature applications of strain gages
High-temperature strain gages and their installation
Commercial Equipment
Rosette analysis and interpretation of observations

STRAIN GAGE LABORATORY COURSE

August 28 - September 1, 1961 (Monday through Friday)

For those who wish to gain personal experience in the installation of gages and the handling of equipment, a second course containing a series of laboratory experiments will be offered. The laboratory program is supplementary to the lecture course. It will consist of a set of projects, each containing one or more experiments, which have been designed to illustrate the topics taken up in the lecture course during the previous week. The following exercises will be included to provide practice in the practical applications of strain gage techniques:

Strain Gage Installation for normal and high temperature applications
Static observations with single and multiple gage bridges
Dynamic strain Measurements
Characteristics and checking of electrical equipment used with strain gages
Strain Rosette Analysis
Characteristics and handling of certain commercial equipment
Measurement of Force, Torque, Bending, and Moments at inaccessible locations

Since the benefits which can be obtained by attendance at the laboratory program depend to a large extent upon an understanding of the underlying theory, registration in the laboratory course will be limited to those who have attended the previous lecture course, or its equivalent in prior years.

INDUSTRIAL PHOTOELASTICITY

(Lectures & Laboratory)

September 5 - 9, 1961 (Tuesday through Saturday)

The course in Industrial Photoelasticity will review the fundamental theory and laboratory techniques related to the classical applications of polarized light for solving stress problems. In addition to this, discussions of some of the most advanced techniques, including Three-Dimensional Photoelasticity and Photoelastic Surface Coatings (PhotoStress), will also be presented.

The course will emphasize the ways in which photoelasticity can be used to solve industrial problems. Attention will also be given to the advantages and limitations of the method and its relation to other experimental techniques with which it may often be combined most effectively.

The material will be presented through lectures, demonstrations, and laboratory exercises. Consideration will also be given to the properties and selection of photoelastic materials as well as the specification of equipment required to set up a laboratory. This program has been designed especially for the engineer or metallurgist who has little or no previous knowledge of the subject.

TUITION

The tuition per registrant for each course will be as follows:—

Strain Gage Lecture Course	**$175.00**
Strain Gage Laboratory Course	**$175.00**
Industrial Photoelasticity	**$200.00**

FOR FURTHER INFORMATION, call or write,

ENGINEERING EXTENSION, DEPARTMENT OF ENGINEERING,
Room 6266, ENGINEERING-PHYSICAL SCIENCES BUILDING,
UNIVERSITY OF CALIFORNIA, LOS ANGELES, 24, CALIFORNIA

Telephone — GRanite 3-0971, Extension 369

Sources of information on

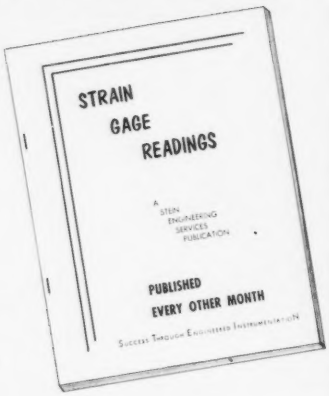
STRAIN GAGES

STRAIN GAGE TECHNIQUES

by W. M. Murray
& P. K. Stein

A comprehensive reference book on basic theory and experiments

669 pages, 8½ x 11, cloth bound, photo-offset from typewritten copy



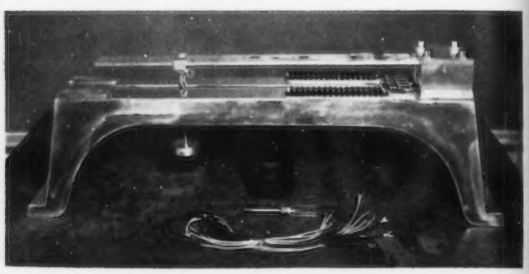
STRAIN GAGE READINGS

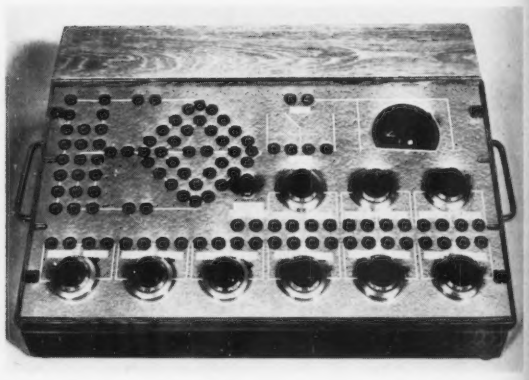
P. K. Stein, Editor

A bi-monthly journal devoted to new applications and techniques and reviews of current literature

THE STRAIN GAGE INSTRUCTOR

Aluminum stand and cantilever beam with mounted strain gages





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EXPERIMENTAL EQUIPMENT

for demonstrating the theory and performing the laboratory experiments which are described in STRAIN GAGE TECHNIQUES and STRAIN GAGE READINGS
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